

OIL BURNERS

OIL BURNERS

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To

ISADORE L. SCHURMAN

whose steadfast faith, counsel, and
encouragement have made this work
possible.

PREFACE

In presenting this volume to the petroleum industry generally and to the oil-burner and fuel-oil divisions thereof especially, the author asks the indulgence of his readers for using a somewhat different method of treatment than has been employed in the previous very scant literature on the subject. The general plan of the text is first to state the fundamental principles from which the subject matter of the chapter is to be built up, and then to proceed rapidly through what would be ordinarily an elaborate development of theoretical material to the pertinent conclusions. Those readers having academic training may feel that the book is unnecessarily prolix in elementary treatment, while those not so fortunate as to have pursued formal courses in technology but who have had intimate and practical association with the industry may find some of the physics, chemistry and mathematics not sufficiently detailed. It has been the hope of the author that the text would strike a middle course.

The book describes basic concepts and essential engineering as well as actual practice. While modes change, engineering develops onward from what is already known and established. Thus it is hoped that what is presented may be as applicable to the oil burner of a decade or two hence as to the products of today's manufacturing. Just as in the case of so many other arts which developed into sciences, as the facts of laboratory investigation and the interpretations of mathematical analysis were logically arranged, so oil burning has now become a branch of engineering. This maturing of the art into the science is the principal inspiration for this book. But in trying to arrange the material in a manner suited to an engineering text, the author has also tried to make the book as practical as possible, so that it might be useful in the routine conduct of any of the technical departments of the industry.

Frequent repetition of text material has been avoided by segregation of classes of apparatus into their individual chapters. Thus accessory equipment having application to many different

types of oil burners will be found in complete treatment in separate chapters; this applies to pumps, controls, pressure regulators and many other forms of equipment which are indispensable adjuncts to many burners and yet cannot be qualified or distinguished as peculiar to one type of burner. It has thus been necessary to describe the accessory only once, and to indicate its applications in as many cases as exist.

The oil-burner industry itself stands independently among American business enterprises, and it has attained that eminence because it has zealously guarded its progress, and because both its leaders and its rank and file have visualized an ideal. That ideal springs from the American spirit of development, tempered by the soundly promulgated program of natural-resource conservation. At the same time it draws inspiration from the pervading spirit of the entire petroleum industry, a spirit which can perhaps best be exemplified by the following excerpt from a publication of the American Petroleum Institute, "Petroleum—The Story of an American Industry":

It is doubted if any other industry ever was forced in so short a time to make such enormous capital outlays, to enter upon such great expansion programs, and to meet such rapidly growing and changing demands. The fact that this country never, even for a day, has been threatened with a serious or generally inconvenient shortage of petroleum or petroleum products, and that the market price of these products has shown through the years an almost uninterrupted downward trend, may be taken as proof of the ingenuity, foresight, business sense, and executive ability of those Americans who pioneered, developed, and expanded this industry.

The author makes grateful acknowledgment of kind cooperation and assistance to many friends and associates who have given liberally of their time, knowledge and experience during the preparation of this book. Among these, especially, are Mr. Paul R. Unger of Mid-West Heat Service Company, Chicago, and Mr. Fred Ravensbeck, Chief Engineer of Ace Engineering Company, Chicago, both of whom have in large measure rendered valuable assistance on many portions of the book. Mr. William Goodman of Trane Company, La Crosse, Wisconsin, gave inspiring aid in many engineering details, particularly on Chap. XX. Others who contributed to preparation or rendered helpful

criticism are Mr. Theodore J. Kauders of Patterson-Kelley Company's Philadelphia Office, who helped prepare the chapter on Preheating; Mr. J. O. Rosche, Chief Engineer of Chalmers Oil Burner Company, Minneapolis, and Mr. O. C. Schroeder, Vice-President and Chief Engineer of Automatic Burner Corporation of Chicago, both of whom aided in writing portions of the Domestic Oil Burner chapter and other equipment for small burners; Mr. A. H. Koch of Minneapolis-Honeywell Regulator Company, Brown Instrument Division, who assisted in writing the discussion of pyrometers in Chap. VIII; Mr. J. Krawetz of Phoenix Chemical Laboratory, Chicago, who very kindly criticized Chaps. I and II; Mr. Erwin Mautner, also of Mid-West Heat Service Company, whose experience contributed to the text on bakery practice; Mr. F. I. Raymond of F. I. Raymond Company and Mr. H. J. Kucera of James P. Marsh Company, both of Chicago and both contributors to the preparation of Chap. XX; Mr. A. Bogot of Marnis Oil Company, Inc., New York, who assisted in the calculation of much of the data in the book, as did also Mr. Robert K. Deutsch, a student at Massachusetts Institute of Technology.

KALMAN STEINER.

CHICAGO,
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INTRODUCTION

While the knowledge and use of petroleum date back almost as far as recorded history itself, we can begin any consideration of its exploitation in America with the discovery of petroleum in the first American well, that of Colonel Drake, in Pennsylvania, in 1859. From that year until 1892 most of the production of petroleum was used in the manufacture of illuminants, lubricating oils, paraffin, etc., although oil was gradually adapted for fuel purposes, particularly in regions near the oil fields. Many types of steam atomizing burners came into use during that period. The Boston and Albany Railroad in 1881 was running an oil-burning locomotive. Beginning about 1880, Best, Ray and Johnson were pioneering oil-burning devices on the Pacific Coast.

The application of fuel oil to central heating plants had its inception in Chicago and much of its subsequent development and growth occurred in that city. The fifty-four boilers used to heat the buildings comprising Chicago's World's Fair in 1893 were oil fired—and were thus the first large installations of fuel-oil burners for the heating of buildings. Chicago's next World's Fair, A Century of Progress, in 1933, beheld a total of 102,500 oil-burner installations in the city's metropolitan district, with 287 firms engaged in the retail sale of oil burners, and some forty firms distributing fuel oil to these installations.

The fair of 1893 made another contribution to a permanent development in Chicago. One of the electrical inspectors, who had been lent by the fire insurance companies to the fair to assist in preventing loss of property by fire and in avoiding accidents of shock, remained in Chicago for the purpose of testing and classifying for the insurance companies of that city electrical appliances, acetylene and gasoline lighting appliances and cooking and heating devices. His reports subsequently received nationwide distribution and acceptance, and they

formed the basis for the organization in 1901 of Underwriters' Laboratories.

This institution made its first listing of an oil-burning system in July of 1912. This listing was withdrawn in August of 1915 owing to discontinuance of manufacture of the device. It had covered a class of burners now designated as industrial. The first listing of a domestic oil burner was issued about July, 1918, when a natural-draft burner was passed for the Baldwin Sales Company. The first listing of a mechanical-draft burner, representing the crystallization of a modern domestic burner, came late in 1919, and was issued for the No-Kol Burner. Thus in 1919 only 1 burner appeared in the Laboratories' published listings—in 1923 there were 6; in 1924, 11; in 1925, 27; in 1936 over 200 burner manufacturers appeared in the list of approvals.

During 1937 fuel oil is being used to heat over one million American homes; and at least sixty thousand installations are in use for heating apartment and hotel buildings, factories, mill and industrial buildings, schools, churches, hospitals and sanitaria and public buildings including office buildings, post offices, governmental buildings, etc. A total of two and a half million boiler horsepower is being developed from oil-fired boilers, in addition, for central stations and industrial plants, and there is also a wide application of oil burners to foundry and smelting work. During 1934 and 1935, the consumption of fuel oil in the United States, according to the U. S. Bureau of Mines, was 346,077,000 and 387,511,000 barrels, respectively. The Bureau divides this consumption into the following classifications:

Uses	1934, bbl. of 42 gal.	1935, bbl. of 42 gal.
Railroads.....	52,581,000	55,651,000
Steamships (including tankers)....	69,262,000	74,581,000
Gas and electric power plants....	23,143,000	23,647,000
Smelters and mines.....	2,682,000	2,448,000
Manufacturing industries.....	54,260,000	61,128,000
U. S. navy, army transports, etc.	7,914,000	10,428,000
Used as fuel by oil companies....	47,404,000	48,116,000
Miscellaneous uses.....	12,253,000	13,133,000
Domestic and commercial heating	60,822,000	76,853,000
Range oil.....	15,756,000	21,526,000

Exact figures are not available for the number of commercial and industrial oil-burner installations, but the following tabulation illustrates the growth of the domestic oil-burner industry.¹

**DOMESTIC BURNERS OPERATING AT THE END OF YEAR AND OUTPUT
BY YEARS**

Year	Conversion		Boiler-burner units*		Furnace burner units *		Total output	Total No. in use
	Output	No. operating	Out-put	No. operating	Out-put	No. operating		
1920		12,500						12,500
1921	9,000	21,500					9,000	21,500
1922	35,300	56,800					35,300	56,800
1923	43,000	99,800					43,000	99,800
1924	79,700	178,700					79,700	178,700
1925	45,200	177,400†					45,200	177,400
1926	77,700	243,100					77,700	243,100
1927	82,700	322,000					82,700	322,000
1928	107,500	424,100					107,500	424,100
1929	131,300	547,500					131,300	547,500
1930	126,400	665,100					126,400	665,100
1931	104,000	760,800					104,000	760,800
1932	86,200‡	839,200		3,200		800	86,200	843,200
1933	80,800‡	907,900	7,800	11,000	600	1,400	89,200	920,300
1934	100,000	991,600	9,300	20,300	800	2,200	110,103	1,014,100
1935	142,350	1,117,850	11,600	31,300	3,350	5,550	156,700	1,154,700

* Boiler-burner and furnace-burner units were sold in small numbers prior to 1932, but lack of accurate data prevents giving output by years prior to 1933.

† The decline in the number of burners in operation in 1925 compared with 1924 is explained by the fact that in 1925 a large producer of natural-draft burners collapsed. Many of the burners sold by this manufacturer were removed in 1925, and 1926, because of lack of service.

‡ The conversion burner output for 1932 includes an undetermined number of boiler- and furnace-burner units. Conversion burner output increased slightly in 1933 compared with 1932.

There were 1,449,100 distillate burners operating at end of 1935.²

The oil-burner industry of this country has done more than merely create a new industry for itself and a new market for petroleum refiners. It has set fuel competition on so high a plane that the coal and gas industries have been forced into

¹ Compiled by *Fuel Oil Journal*.

² Estimate by Oil Burner Institute.

adopting modern methods of automatic combustion and heating control and into attaining new heights of efficiency in fuel utilization. Thus this newcomer amongst long-established fuels has definitely assisted the country's program of fuel conservation.

This work is not intended to be or thought to be a duplication of technical matter which can be found or is available elsewhere. It is intended to present the technology of an important division of American industry. The literature of the petroleum industry is a vast one, just as is the industry itself. Yet nowhere in this library is there existent an adequate treatment of fuel oil and oil burning. It is the aim of this book to give an exhaustive and comprehensive treatment to this subject, so that it may serve both as a reference book to the industry at large and as a text to those workers in the field who seek broader information on the many ramifications of the subject.

OIL BURNERS

CHAPTER I

HYDROCARBONS, PETROLEUM AND PETROLEUM REFINING

In the physical universe in which we live, three types of matter or substance have been recognized by chemists—elements, compounds and mixtures of these two. Physicists also have classified matter into three groups—solids, liquids and gases. It happens that any one of the chemical types of matter can exist in any one of the three physical forms, *i.e.*, as solid, liquid or gas; similarly any one of the three physical phases possible may represent any one of the three chemical classes, *viz.*, element, compound or mixture.

These scientific descriptions of matter are extremely useful, and any one dealing in or concerned with scientific or technical work will find that the ability to distinguish these groups each from the others, or to recognize the physical and chemical significance of the classification of a substance, will help considerably in comprehending the application of substances. Thus, to make even an elementary study of fuels and their combustion, the student should be able to think of fuels in terms of these simple scientific concepts. These six terms will be defined in as simple a manner as will permit of their proper recognition.

An element is a substance which cannot be resolved into other substances, different from itself, by ordinary chemical or physical processes: The smallest possible particle of an element is an atom, but an atom is a unit so small as to be far below the range of the most powerful microscope. There are about ninety elements, of which the most common are oxygen, nitrogen, carbon, hydrogen, iron, copper, chlorine, sodium, etc., their respective chemical symbols being O, N, C, H, Fe, Cu, Cl, Na. The lightest of the elements is hydrogen; on an arbitrary scale, its atomic weight is 1.008; on the same scale, the atomic weights of the

others just named are oxygen, 16; nitrogen, 14.008; carbon, 12; iron, 55.84; copper, 63.57; chlorine, 35.46; sodium, 23. The atomic weight of an element therefore signifies its relative weight compared to the others. The entire scale is based upon the atomic weight of oxygen at 16.

In conformity with definite chemical laws, most of the elements combine with other elements to form chemical compounds, which are substances possessing properties entirely different from the elements composing them. A familiar chemical compound is ordinary table salt, sodium chloride; its chemical formula, NaCl , indicates that one atom of sodium has united with one atom of chlorine to form a molecule of another substance, sodium chloride. The smallest particle of a compound is a molecule, and like the atoms which have united to form it, a molecule is infinitesimally small and not visible under the microscope. Other familiar compounds are water, H_2O ; carbon dioxide, CO_2 ; iron rust or iron oxide, Fe_2O_3 ; nitric acid, HNO_3 . The subscript indicates the number of atoms of the element preceding the number which occur in the compound; the absence of a number signifies that a single atom of the element is present.

It therefore follows that every pure substance is either an element or a compound. Elements and compounds may, however, be present in the same physical form as a mixture. In the case of a mixture, there is no chemical union of the components, and very often the components may be separated from each other without chemical means. Thus a mixture of sugar and sand, which are pure substances in themselves, may be separated physically by dissolving the sugar in water. Since sand is insoluble, it can be filtered out of the sugar solution; the sugar then may be recovered by allowing the water to evaporate. In this manner the two substances, sugar and sand, are separated from each other without chemical change. A solution may be considered a special type of mixture.

A solid may be defined as matter which resists, up to certain limits, forces which tend to deform it or to change its shape. Solids which are pure substances, *i.e.*, either element or compound, have definite melting points. Other solids may be mixtures of substances and may not have definite melting points. Some solids may be quite complex in their chemical composition, as for instance wood, which undergoes various chemical changes

upon being heated. Many solids have definite crystalline forms which they assume upon solidifying from either the liquid state or from solution. Other solids, usually complex mixtures, such as coal, may be amorphous, or lacking in crystalline form.

Liquids may be conceived of as matter possessing definite volume but lacking definite shape. Liquids simply assume the form of the containers holding them, and their upper surfaces if undisturbed become horizontal planes. A liquid, if a pure chemical substance, has a definite freezing point; this freezing point is lowered by the presence in solution of another substance, and upon freezing, the liquid usually separates from the dissolved substance. Similarly, a pure substance has a definite boiling point; but this boiling point is increased by the presence in the liquid of dissolved matter, the amount of the increase in boiling point being proportional to the concentration of the dissolved matter.

A gas has the property of completely filling, in a uniform distribution, the space which confines it. Hence a gas has neither definite shape nor definite volume. All liquids upon being heated to their boiling points change into the vapor or gaseous phase. The terms vapor and gas are quite similar, although usually vapor refers to the molecules of a liquid which have evaporated from the liquid surface below the boiling point, while gas designates a state of a substance above the boiling point. Steam is generally considered to be a vapor when at or just above its boiling temperature, and to be a gas when at temperatures considerably above the boiling point. Air likewise is a gas but is not a pure substance; it is a mixture of gases, chief of which are the two elementary gases oxygen and nitrogen. A given quantity of any gas can be established quantitatively only if referred to by weight, or by temperature, pressure and volume. Although gases may have widely varying chemical characteristics, all gases conform closely to definite changes in volume with temperature and pressure.

Since liquids and gases collectively are materials which can be measured by flow, they are often referred to as fluids. The hydraulic laws which govern their flow through conduits are in many respects similar for both. Gases and liquids are also in close agreement with each other in respect to many aspects of their physical behavior.

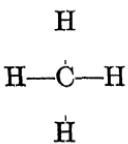
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HYDROCARBONS

With these simple scientific concepts in mind, it is possible to proceed directly to a consideration of the chemical composition of the substances which constitute petroleum. These are a group of chemical compounds known as hydrocarbons, and as the name implies, they contain only two elements, carbon and hydrogen.

The simplest known substance in the hydrocarbon group is a compound containing one carbon and four hydrogen atoms, CH_4 . This substance is called methane. Methane is a gas, widely encountered in fuels. Natural gas consists of 90 to 95 per cent methane; manufactured fuel gas contains it in varying amounts; coal gas, which is formed by distillation of coal during the coking process, contains large amounts of methane. It is formed during the decomposition of vegetable matter in marshes and hence is sometimes known as marsh gas. It occurs in coal mines, and miners have called it mine damp.

It is convenient to assign a definite arrangement to the atoms which constitute a molecule of a substance. For various proved chemical reasons, the arrangement for methane shows the carbon atom to be central with the four hydrogen atoms attached directly to the carbon:

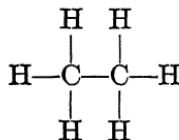


The bond shown between the carbon atom and each hydrogen atom is the representation of a property called valence. Valence may be defined as the combining capacity of an element. According to this definition, and from the structure shown above for methane, it is seen that carbon exhibits a valence of four and hydrogen a valence of one. Throughout the entire field of chemistry, hydrogen always exhibits this valence of one, while carbon usually possesses a maximum valence of four. Carbon, however, in common with some other elements, may under certain conditions display a valence lower than its maximum.

It is usually possible to determine the valence of an element in a substance from the chemical formula. Thus, since the formula of water is H_2O and since it is known that the valence of hydrogen

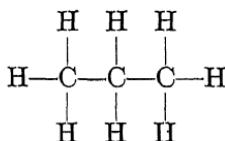
is one, it follows that the valence of oxygen must be two, since it can combine with two hydrogens. The formula of carbon dioxide is CO_2 , which indicates that here also carbon has a valence of four, since two oxygens have a total valence of four. But in CO , carbon monoxide, it appears that the valence of carbon has been reduced to two. A single carbon atom has never been known to combine with less than four hydrogens, but in more complex hydrocarbons, carbon sometimes has a lower valence.

Methane is the first member of a series of hydrocarbons, the next member of which is ethane, C_2H_6 , represented structurally as



Here again carbon shows a valence of four, since both carbon atoms are bonded to four other atoms—three hydrogens and one carbon. Ethane is also a gas, occurring in small quantities in natural gas and manufactured fuel gases.

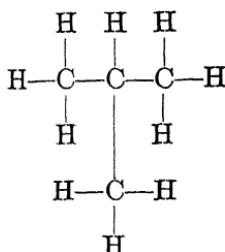
In ethane is seen the first manifestation of a tendency which carbon exhibits as a pronounced chemical property, *viz.*, to form chains or compounds in which numerous carbons are attached to each other consecutively. The next member of this series would contain three carbons, and eight hydrogens; it is known as propane, is also a gas, the formula is C_3H_8 , and the structure is



The series of which methane, ethane and propane are the first three members is a long one. The general formula for any member is $\text{C}_n\text{H}_{2n+2}$, in which n represents the number of carbon atoms. In petroleum there have been found members of this series as high as $\text{C}_{60}\text{H}_{122}$. The members of this group are known as the paraffins and also as the methane series.

These hydrocarbons also form compounds in which the carbons do not adhere strictly to a straight-chain formation, but show side branches, such as

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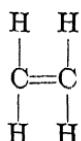


which is called isobutane. Normal butane would have four carbon atoms in a straight chain, but both forms of butane would be represented as to composition by C_4H_{10} .

The paraffin hydrocarbons are characteristically inactive chemically and can be made to enter into chemical reactions only when acted upon by powerful reagents, such as strong nitric or sulphuric acid, and then only when long periods of contact and admixture are allowed. To this chemical inertness the hydrocarbons owe their name, which is derived from the Latin *parum affinis* (small affinity). As the number of carbon atoms increases, the compounds manifest progressive changes in physical properties, such as the change from gases to liquids to solids. The table on p. 7, gives the names, formulas and some physical data of several normal paraffins.

An entirely separate series of hydrocarbons exists having the general formula C_nH_{2n} . This is known as the olefins or ethylene series, since ethylene, C_2H_4 , is the first member. The analogue of methane, which in this series would be methylene, has never been prepared or found in natural occurrence.

A study of the chemical reactions of ethylene and its derivatives has resulted in there being assigned to ethylene the structural formula



in which a double bond is shown between the two carbon atoms. This gives to each carbon a valence of four, but it would seem that this is an unsaturated condition of the molecule and the presence of such a peculiar bond between the carbons would permit of easy entrance of other substances into the compound.

Such is actually the case. For instance, when chlorine is brought into contact with ethylene, an immediate reaction occurs in which two chlorine atoms enter the molecule to form dichlorethane, $\text{ClCH}_2\cdot\text{CH}_2\text{Cl}$, also called ethylene chloride.

The next member of the olefin series is propylene, C_3H_6 . Its structural formula is $\text{CH}_3\cdot\text{CH}=\text{CH}_2$, again exhibiting the double

TABLE I.—TYPICAL NORMAL PARAFFIN HYDROCARBONS

Name	Formula	Molecular weight	Melting point, °F.	Boiling point, °F.	Specific gravity at temp. shown, °F.*
Methane.....	CH_4	16.03	-296.5	-258.5	0.415/-263
Ethane.....	C_2H_6	30.05	-277.6	-126.9	0.5459/-127
Propane.....	C_3H_8	44.06	-309.9	-44.1	0.5853/-48.1
Butane.....	C_4H_{10}	58.08	-211.0	31.1	0.60/32
Pentane.....	C_5H_{12}	72.09	-201.8	96.8	0.6214/77
Hexane.....	C_6H_{14}	86.12	-139.7	155.7	0.65502/77
Heptane.....	C_7H_{16}	100.12	-131.2	209.1	0.67963/77
Octane.....	C_8H_{18}	114.14	-70.4	258.1	0.6988/77
Nonane.....	C_9H_{20}	128.16	-58.0	303.3	0.71398/77
Decane.....	$\text{C}_{10}\text{H}_{22}$	142.17	-21.6	345.2	0.72643/77
Undecane.....	$\text{C}_{11}\text{H}_{24}$	156.20	-14.8	383.5	0.73667/77
Duodecane.....	$\text{C}_{12}\text{H}_{26}$	170.22	10.4	421.5	0.74542/77
Tridecane.....	$\text{C}_{13}\text{H}_{28}$	184.24	21.2	453.2	0.7608/59
Tetradecane.....	$\text{C}_{14}\text{H}_{30}$	198.25	41.0	486.5	0.766/68
Pentadecane.....	$\text{C}_{15}\text{H}_{32}$	212.26	50.0	518.9	0.7689/68
Hexadecane.....	$\text{C}_{16}\text{H}_{34}$	226.27	64.5	549.5	0.7751/68
Heptadecane.....	$\text{C}_{17}\text{H}_{36}$	240.28	71.6	563.9	0.822/59.9†
Octadecane.....	$\text{C}_{18}\text{H}_{38}$	254.30	82.4	602.6	0.7768/82.4
Nonadecane.....	$\text{C}_{19}\text{H}_{40}$	268.32	89.6	626.0	0.7774/89.6
Eicosane.....	$\text{C}_{20}\text{H}_{42}$	282.34	98.9	243.5‡	0.837/89.6‡
Tricosane.....	$\text{C}_{23}\text{H}_{48}$	325.38	118.4	609.3	0.7799/118.4
Pentacosane.....	$\text{C}_{25}\text{H}_{52}$	352.41	129.2	306.5‡	
Nonocosane.....	$\text{C}_{29}\text{H}_{60}$	398.48	145.4	354.2‡	
Duotriacontane.....	$\text{C}_{32}\text{H}_{66}$	450.53	158.0	393.8‡	
Pentatriacontane.....	$\text{C}_{35}\text{H}_{72}$	492.58	167.0	431.6‡	0.846/80.†

* Specific gravity of the liquid.

† Data from Roy Cross, "Handbook of Petroleum, Asphalt and Natural Gas," Kansas City Testing Laboratory. All other data from "Chemical Engineers' Handbook," McGraw-Hill Book Company, Inc.

‡ *In vacuo*.

bond which indicates an unsaturated compound. No matter how long the chain becomes in higher members of this series, the two carbon atoms which each lack one hydrogen atom of the analogous paraffin are always adjacent in the molecule. The olefins, as the paraffins, also form compounds with side chains which are typical iso-compounds. And as the chain becomes longer, the olefins show the same progressive changes from gases

to liquids, and from liquids to solids, as do their paraffinic homologues. The following table lists some olefins, with physical data:

TABLE II.—TYPICAL OLEFIN HYDROCARBONS

Name	For-mula	Molec-ular weight	Melting point, °F.	Boiling point, °F.	Specific grav-ity at temp. shown, °F.*
Ethylene.....	C ₂ H ₄	28.03	-272.9	-155.0	0.3384/32
Propylene.....	C ₃ H ₆	42.05	-300.8	-52.6	0.6095/-52.6
Butylene, α	C ₄ H ₈	56.06	-202.0	21.0	0.668/32
Amylene, i - β	C ₅ H ₁₀	70.08	-191.2	101.1	0.668/55.4
Hexylene.....	C ₆ H ₁₂	84.10	-145.3	154.4†	0.683/68‡
Heptylene.....	C ₇ H ₁₄	98.11	208.4†	0.703/66.2‡
Octylene.....	C ₈ H ₁₆	112.13	257.0	0.722/68
Nonylene.....	C ₉ H ₁₈	126.14	307.4†	0.743/68†
Decylene.....	C ₁₀ H ₂₀	140.16	341.6‡	0.763/32‡
Pentadecylene.....	C ₁₅ H ₃₀	210.24	476.6†	
Eicosylene.....	C ₂₀ H ₄₀	280.32			
Melene.....	C ₃₀ H ₆₀	420.48	143.6‡	0.890†

* Specific gravity of the liquid.

† Data from Roy Cross, "Handbook Petroleum, Asphalt and Natural Gas," Kansas City Testing Laboratory.

‡ Data from Chemical Rubber Company, "Handbook of Chemistry and Physics." All other data from "Chemical Engineers' Handbook," McGraw-Hill Book Company, Inc.

As a class, the members of the ethylenes are much more active chemically than the paraffins. As mentioned above, this is due to the unsaturated condition of the two carbon atoms sharing the double bond, for it is always at these two atoms that substitution or addition products are formed. The presence of unsaturated compounds in liquid fuels is of special importance, and further reference will be made to them in the consideration of the properties of fuel oils.

A still higher degree of unsaturation is displayed by another series of hydrocarbons of which acetylene is the first member. Acetylene has the composition formula C₂H₂, and it and higher members have the general formula C_nH_{2n-2}. The structure of acetylene, as interpreted from its chemical behavior, can best be represented by the use or conception of a triple bond between the two carbons, HC≡CH. The homologue of propane is propyne or allylene, CH₃C≡CH. As a group, this series is not long,

acetylene being the most prominent and most useful. Others are shown below.

TABLE III.—TYPICAL ACETYLENE HYDROCARBONS

Name	For- mu- la	Molec- ular weight	Melting point,	Boiling point,	Specific gravity at temp. shown, °F.*
Acetylene.....	C ₂ H ₂	26.02	-115.2	-118.5	0.6208/-119.2
Allylene.....	C ₃ H ₄	40.05	-166.0†	-10.3†	0.678/-16.6‡
Crotonylene.....	C ₄ H ₆	54.05		80.6‡	
Valerylene.....	C ₅ H ₈	68.06		131.0‡	
Isopropyl acetylene...	C ₅ H ₈	68.06		158.0‡	0.685/32‡
				83.0†	
Butyl acetylene.....		82.08		181.4‡	
Heptine.....	C ₇ H ₁₆	96.1		210.2‡	0.738/53.6

* Specific gravity of the liquid.

† Data from Chemical Rubber Company, "Handbook of Chemistry and Physics."

‡ Data from Roy Cross, "Handbook of Petroleum, Asphalt and Natural Gas," Kansas City Testing Laboratory. Other data from "Chemical Engineers' Handbook," McGraw-Hill Book Company, Inc.

There exists another group of hydrocarbons of the same degree of unsaturation as the acetylene series, but lacking a triple bond. These compounds are said to be isomeric with the acetylenes,¹ and they are called the diolefins. A study of their chemical properties indicates the presence of two double bonds. Thus allene, C₃H₄, has the structure H₂C=C=CH₂, and is isomeric with allylene, C₃H₄. Of particular interest in the diolefin series is the substance isoprene, C₅H₈, which is partially converted into rubber when subjected to heat and pressure. Isoprene itself is a liquid whose boiling point is 94.1°F. Other members of the series are described in the table on p. 10.

The compounds thus far discussed have all been of the so-called chain hydrocarbon type, *i.e.*, the carbon atoms stretch out one after the other, with the exception of the side branches of the iso-forms, to make a carbon chain. These chain hydrocarbons are known collectively as aliphatic compounds, for the reason, as

¹ Substances are said to be isomeric, or one to be an isomer of the other, when they have identical constitutional formulas but different structural formulas. That is to say, they are different substances but are composed of the same elements in the same proportions.

will be shown later, that acids which are derived from chain hydrocarbons are the acids which form fats and oils and greases.

Besides the vast group of chain compounds which are known to aliphatic chemistry, there is another entirely different group in which the typical hydrocarbon is a cyclic compound. In these cyclic arrangements, the carbon atoms form rings or closed

TABLE IV.—DIOLEFIN HYDROCARBONS

Name	For- mu- la	Molec- ular weight	Melting point,	Boiling point,	Specific gravity at temp. shown, °F.*
Allene.....	C ₃ H ₄	40.05			
Butadiene.....	C ₄ H ₆	54.05	23.4	0.650/21.2	
Isoprene.....	C ₅ H ₈	68.06	-184.0	94.1	0.6806/68
Trimethyl butadiene.....	C ₇ H ₁₂	96.1		197.6	0.723/68†

* Specific gravity of the liquid.

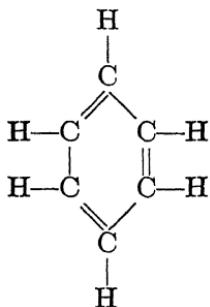
† Data from Roy Cross, "Handbook of Petroleum, Asphalt and Natural Gas," Kansas City Testing Laboratory. Other data from "Chemical Engineers' Handbook," McGraw-Hill Book Company, Inc.

chains, and the number of carbons in the ring does not usually attain the enormous value of some aliphatic substances. Thus by far the most common type of cyclic arrangement has six carbons in the ring, although rings of from three to nine carbon atoms are known.

The hydrocarbon family which originates from the six-carbon ring is the most important group known to science and the most extensive and diverse in number and properties. The class name of aromatic has been assigned to these substances. The reason probably was that most of the substances found in nature with an agreeable odor were originally thought to belong to this group; although it is now known that some aliphatic compounds can also have agreeable odors while many aromatics are pungent and sharp and disagreeable in odor. But the derivatives of aromatic hydrocarbons are found in a great many seemingly unrelated things, as dyes and carbolic acid, motor benzol and perfumes, T.N.T. and moth balls.

The fundamental compound of the aromatics is benzene, C₆H₆, just as the fundamental compound of the aliphatics is methane. Just as any aliphatic substance may be considered as derived

from methane by the replacement of one or more of its hydrogens by other atomic structures, so aromatic substances may be considered as derived from benzene by the replacement of one or more hydrogens by other groups. The structure most generally assigned to benzene is



This gives to the benzene molecule three double bonds. It was noted in the case of aliphatics that a double bond was an indication of increased chemical activity, so it could be expected that benzene would be more active than methane. In addition, the ratio of hydrogen to carbon is much lower than with the paraffins, consequently benzene would be classified as an unsaturated compound if it were not that its structural formula is symmetrical. Actually, benzene is much more active than methane, and readily reacts with a great many other substances to form substitution products.

It will be remembered that a hydrogen atom of methane could be replaced by another methyl group to form the next higher hydrocarbon, ethane. In a similar way a hydrogen atom of benzene can be replaced by a methyl group to form $C_6H_5CH_3$, which is toluene. If two hydrogens of benzene are replaced by methyl groups, the result is a xylene, of which there are three, differing in the relative positions of the hydrogens which have been replaced on the benzene ring. The possible number of derivative hydrocarbons from benzene is thus seen to be enormous, for with only a single benzene atom, it is possible to replace, singly or more than one at a time, each of six hydrogens with any of hundreds of aliphatic hydrocarbons, while if more than one benzene nucleus is involved in a single molecule, the possible arrangements are, of course, vastly greater. So with this brief description, the aromatics are here left; anyone seeking

more knowledge concerning them may refer to standard works on organic chemistry. A few of them are listed below.

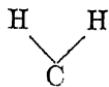
TABLE V.—TYPICAL AROMATIC HYDROCARBONS

Name	Formula	Molecular weight	Melting point, °F.	Boiling point, °F.	Specific gravity at temp. shcwn, °F.*
Benzene (benzol)....	C ₆ H ₆	78.05	42.01	176.22	0.8724/77
Toluene (toluol)....	C ₆ H ₅ CH ₃	92.06	-139.0	167.1	0.8623/77
Xylene (ortho-)....	C ₆ H ₄ (CH ₃) ₂	106.08	-16.8	288.9	0.8745/68
Ethyl benzene.....	C ₆ H ₅ C ₂ H ₅	106.08	-137.9	277.1	0.8669/68
Mesitylene (1, 3, 5-)	C ₆ H ₃ (CH ₃) ₃	120.09	-62.9	328.2	0.8634/68
Dibenzyl.....	C ₆ H ₅ (CH ₂) ₂ C ₆ H ₅	182.11	126.5	544.8	0.9636/122
Triphenylmethane..	(C ₆ H ₅) ₃ CH	244.12	198.5	678.6	1.0166/203

* Specific gravity of the liquid.

All data from "Chemical Engineers' Handbook," McGraw-Hill Book Company, Inc.

Another important series of cyclic compounds composed of carbon and hydrogen are the so-called naphthalenes. Here the most striking fact seems to be that the number of carbons in the ring is half the number of hydrogen atoms, so that the general formula is C_nH_{2n}. Hence the naphthalenes are isomeric with the olefins. But while the olefins exhibit the marked chemical activity of an unsaturated type, the naphthalenes manifest no such activity. In fact the naphthalenes are typically inert and indifferent to the presence of all but the most powerful chemicals, and so are akin in properties to the saturated paraffins. The structure of naphthalene shows this saturated condition. For example, the simplest naphthalene, cyclopropane, has the structure



which gives to each carbon its full valence of four. The next three members of the series are, respectively, cyclobutane, cyclopentane and cyclohexane. Structurally, they are

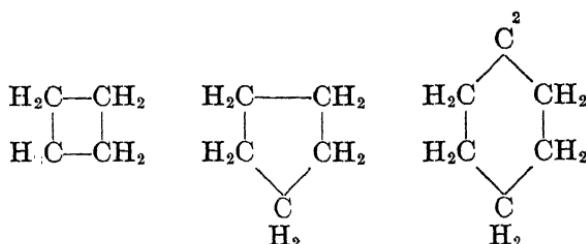


TABLE VI.—TYPICAL NAPHTHENES (POLYMETHYLENES)

Name	Formula	Molec- ular weight	Melt- ing point, °F.	Boiling point, °F.	Specific gravity at temp. shown, °F.*
Cyclopropane (tri-methylene).....	$(CH_2)_3$	42.05	-195.9	47.9	0.720/-110.2
Cyclobutane.....	$(CH_2)_4$	56.06	-58.0	55.4	0.703/32
Cyclohexane.....	$(CH_2)_6$	84.09	43.5	177.4	0.7739/77
Methylcyclohexane	$(CH_2)_5CHCH_3$	98.09	-195.5	214.4	0.7647/77
Cyclopentane.....	$(CH_2)_5$	70.08	122.0	0.750/68†
Cycloheptane.....	$(CH_2)_7$	98.11	10.4	244.6	0.811
Cyclo-octane.....	$(CH_2)_8$	112.13	294.8†	0.840†
Cyclononane.....	$(CH_2)_9$	126.14	339.8†	0.785†

* Specific gravity of the liquid.

† Data from Roy Cross, "Handbook of Petroleum, Asphalt and Natural Gas," Kansas City Testing Laboratory.

All other data from "Chemical Engineers' Handbook," McGraw-Hill Book Company, Inc.

DERIVATIVES OF HYDROCARBONS

It is necessary that a brief study be made of the derivatives of hydrocarbons in order that later it will be possible to discuss fuel oil and its combustion. The formation of derivative substances occurs in fairly regular manner, so that it is generally only necessary to study typical cases. Some of the most important substances reacting with hydrocarbons are the halogens (fluorine, chlorine, bromine and iodine), sulphur, oxygen; simple acids, such as nitric, hydrochloric, sulphuric, acetic; and alkalies, such as sodium hydroxide. In the following discussion of substitution products, however, it is *not* to be assumed that the *chemistry of the reactions or formations is being given*. At most, the text points out the *results* of certain chemical changes, *not the mechanism of those changes*.

Under certain conditions ethyl alcohol, $\text{CH}_3\text{CH}_2\text{OH}$, reacts with hydrochloric acid, HCl, to form ethyl chloride, $\text{CH}_3\text{CH}_2\text{Cl}$, and water, HOH. Ethyl chloride was at one time used extensively as a refrigerant, *i.e.*, as a gas which could be readily compressed, liquefied and again evaporated to perform the refrigerating cycle. Ethyl alcohol is, of course, usually manufactured by fermentation of grains and is seldom made synthetically. The above reaction for the formation of ethyl chloride is typical of the formation of other halides, *i.e.*, if hydrobromic acid had been used, the result would have been ethyl bromide, etc. If chlorine is introduced into the benzene ring, in place of a hydrogen, the compound formed is called chlorbenzene.

It has already been noted above that the presence of an OH group in place of a hydrogen in an aliphatic hydrocarbon creates an alcohol. A similar replacement in an aromatic, such as $\text{C}_6\text{H}_5\text{OH}$, gives a phenol, this one being known also as carabolic acid. As was just indicated, OH may be considered as being derived from water, H_2O , by separating one hydrogen from it.

A typical sulphur compound is one in which the oxygen, as in alcohol, is replaced by a sulphur atom. Such a substance is called a mercaptan, and in the case of the ethyl derivative would be called ethyl mercaptan, $\text{C}_2\text{H}_5\text{SH}$. This particular chemical has an extremely foul odor; as small an amount of it as 2 billionths (0.000000002) of a milligram may be detected by its smell. Mercaptans are encountered in liquid fuels, and in such cases, as can be well imagined, a disagreeable odor is characteristic of the fuel.

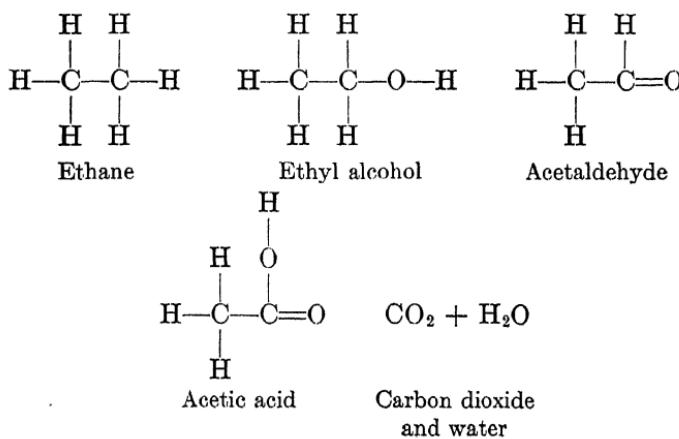
It has already been mentioned that benzene readily reacts with a great many substances. When mixed with nitric acid, which is HNO_3 (structurally $\text{HO}\cdot\text{NO}_2$), in the presence of sulphuric acid the HO group of the acid combines with an H of benzene to form water, while the NO_2 enters the benzene ring and gives nitrobenzene. In a similar manner benzene enters into reaction with sulphuric acid, H_2SO_4 , and yields a sulphonic acid, $\text{C}_6\text{H}_5\text{SO}_3\text{H}$.

A great group of compounds exists which is known as the organic acids. Such an acid always contains the identifying group of atoms —COOH. Thus formic acid, the simplest, is HCOOH . The next in this series would be acetic acid,

CH_3COOH ; this is the acid contained in vinegar. The higher aliphatic acids are commercially important because the soaps are derived from them; the H of the COOH is replaced by a metal, usually sodium, as in sodium stearate, $\text{C}_{17}\text{H}_{35}\text{COONa}$. The simplest of the aromatic acids is benzoic, $\text{C}_6\text{H}_5\text{COOH}$.

Because of its importance in all combustion reactions, the effect of introducing oxygen into hydrocarbon molecules is worth noting. The paraffin radicals, such a methyl and ethyl, combine with oxygen to form what might be called oxides, such as $(\text{CH}_3)_2\text{O}$. Compounds of this type are called ethers. The one just mentioned is dimethyl ether. The ethyl homologue, $(\text{C}_2\text{H}_5)_2\text{O}$, diethyl ether, is the familiar ether used in medicine as an anaesthetic. It is possible for aliphatic and aromatic radicals to enter into the same compound to form an ether such as anisol, or phenyl methyl ether, $\text{C}_6\text{H}_5\text{OCH}_3$. Similarly, phenetol is phenyl ethyl ether, $\text{C}_6\text{H}_5\text{OC}_2\text{H}_5$.

It is well known that the process of combustion is a rapid union with oxygen. While most organic substances do burn, it is possible to add oxygen into a molecule at a regulated rate so that substances other than the products of combustion are formed. To illustrate, the hydrocarbon ethane will be taken through a series of successive oxidations:



Thus though the ultimate products formed are the same as though the original body burned immediately, it is seen that numerous intermediate substances can be formed by oxidation on a slower scale than complete combustion. This is of only

theoretical interest insofar as the combustion of fuels is concerned, but it will be helpful to have this in mind when the chemistry of combustion is discussed.

A summary of the simpler derivatives of hydrocarbons is shown in the following tabulation of three paraffins and two aromatics with their substituents in five cases each.

TYPICAL HYDROCARBONS AND THEIR DERIVATIVES

Name of hydro-carbon	Type of derivative				
	Alkyl chloride	Alcohol	Acid	Ester	Ether
Methane CH_4	Methyl chloride CH_3Cl	Methyl alcohol CH_3OH	Formic HCOOH	Methyl acetate $\text{CH}_3\text{COOCH}_3$	Dimethyl $(\text{CH}_3)_2\text{O}$
Ethane C_2H_6	Ethyl chloride $\text{C}_2\text{H}_5\text{Cl}$	Ethyl alcohol $\text{C}_2\text{H}_5\text{OH}$	Acetic CH_3COOH	Ethyl acetate $\text{CH}_3\text{COOC}_2\text{H}_5$	Diethyl $(\text{C}_2\text{H}_5)_2\text{O}$
Propane C_3H_8	Propyl chloride $\text{C}_3\text{H}_7\text{Cl}$	Propyl alcohol $\text{C}_3\text{H}_7\text{OH}$	Propionic $\text{C}_2\text{H}_5\text{COOH}$	Propyl acetate $\text{CH}_3\text{COOC}_2\text{H}_5$	Dipropyl $(\text{C}_3\text{H}_7)_2\text{O}$
Benzene C_6H_6	Chlorbenzene $\text{C}_6\text{H}_5\text{Cl}$	Phenol $\text{C}_6\text{H}_5\text{OH}$	Benzoic $\text{C}_6\text{H}_5\text{COOH}$	Phenyl acetate $\text{CH}_3\text{COOC}_6\text{H}_5$	Phenyl $(\text{C}_6\text{H}_5)_2\text{O}$
Toluene $\text{C}_6\text{H}_5\text{CH}_3$	Chlortoluene $\text{C}_6\text{H}_5\text{CH}_2\text{Cl}$	Benzyl alcohol $\text{C}_6\text{H}_5\text{CH}_2\text{OH}$	Phenylacetic $\text{C}_6\text{H}_5\text{CH}_2\text{COOH}$		

PETROLEUM

Petroleum is a naturally occurring, viscous and combustible liquid found in various of the earth's upper strata or rock layers. Being classified chemically as an organic substance, petroleum, technically, cannot be called a mineral, since this latter term is reserved scientifically to naturally occurring elements or compounds which have a crystalline structure in the solid state and a definite chemical composition. Petroleum is a mixture of many chemical compounds, some simple, others highly complex. It is popularly, however, considered a mineral, being known variously as rock oil, mineral oil, or crude oil. The word itself is derived from two Latin words: *Petra* (rock) and *oleum* (oil). In the solid state, when frozen, petroleum is waxlike, and more or less crystalline.¹

¹ A.S.T.M. defines petroleum as "a naturally-occurring mixture consisting predominantly of hydrocarbons, or of sulphur, nitrogen, or oxygen deriv-

Asphalt is a solid or semisolid of a similar nature to petroleum; natural deposits of it occur at various points on the earth's surface, and many petroleums leave an asphalt residue when subjected to distillation. The chemical nature of petroleum and asphalt is discussed more fully later in this chapter.

Numerous theories have been advanced to explain the natural formation of petroleum. These hypotheses need not be here considered in detail; standard works on petroleum can be consulted by anyone seeking broader information on the subject. Briefly, these theories hold forth three classes of opinion: inorganic origin, marine-plant origin and marine-animal origin.

The theory of Mendeléeff seems the most plausible of the inorganic group. It holds that there existed in the earth deposits of metallic carbides, such as iron carbide, and that underground waters came in contact with them, thereby forming hydrocarbons just as acetylene is formed by the action of water on calcium carbide. Although some experimental results indicate that hydrocarbons can be so produced there is as yet no evidence to prove that such extensive deposits of carbides ever existed.

There is strong evidence to indicate that much petroleum has been formed from huge accumulations of marine-vegetal matter. When decaying deposits of this material are examined, methane is observed to be present together with oil. Thus petroleum is found where seaweed is decomposing in sand at the present time.

Other petroleum formations are undoubtedly due to decomposition of small aquatic animals. The albuminous and protein matter originally constituting these marine animals would introduce large quantities of nitrogen into such decaying masses, which would account for the high percentages of nitrogen found in such petroleums as California crude. Of course, it is possibly far-fetched to attempt to base theories of formation upon present-day composition of petroleums since the chemical composition would undoubtedly have changed as the oil deposits under the earth's surface migrated in response to geologic changes.

Oil pools or petroleum deposits in the United States are grouped into seven major areas for purposes of reference and designation. A brief description of each follows, and a tabulation is given

tives of hydrocarbons, which is removed from the earth, in liquid state or is capable of being so removed." Report of Subcommittee XX of Committee D-2, Nov. 2, 1933.

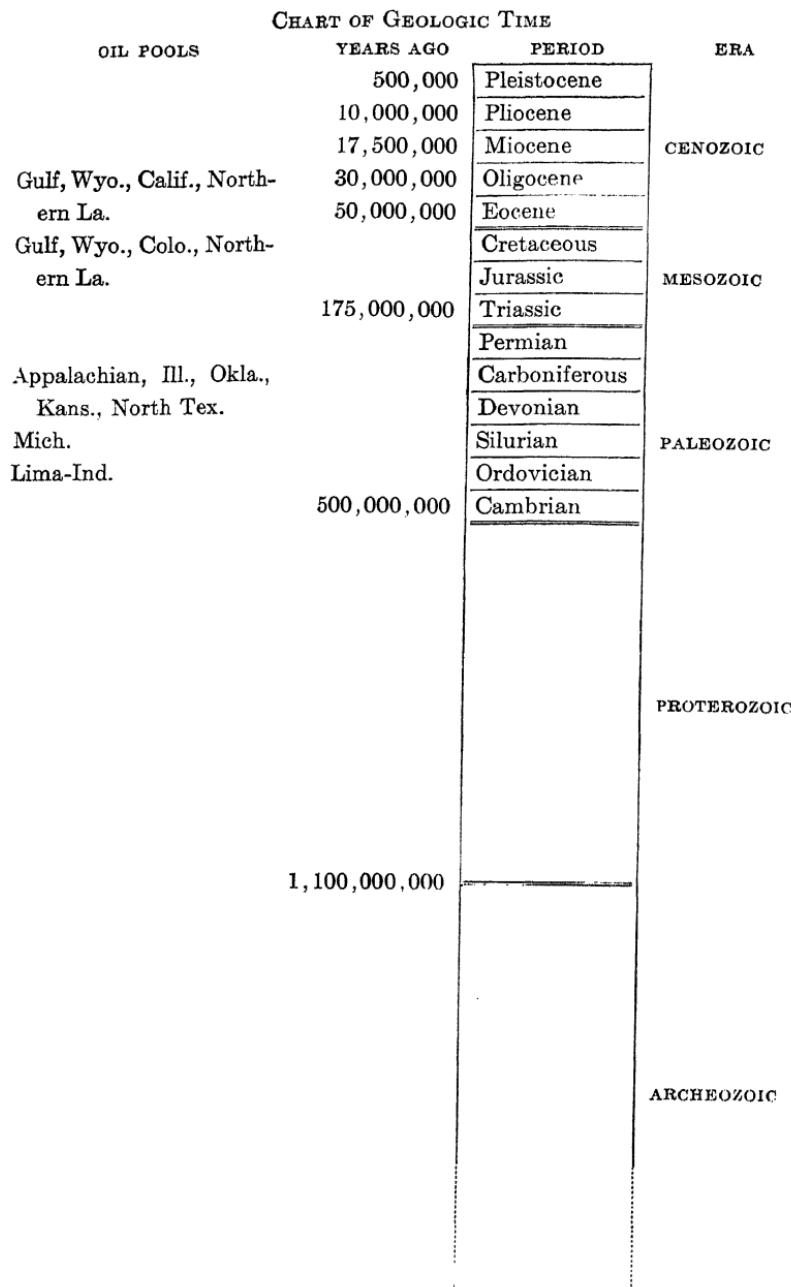


FIG. 1.—Chart of geologic time. On the European system. *Huxley, and Wells, "Science of Life."*) (After Wells,

showing the principal characteristics of the crudes from each area.

In the Appalachian field are included all pools lying east of central Ohio—a north-south area extending from New York to Tennessee, but having its greatest deposits in western Pennsylvania and northern West Virginia. Smaller deposits occur in New York, eastern Ohio, Kentucky and Tennessee. The typical oils of this district are paraffin base, and although they range in color from black to light amber, various shades of green predominate. This is the oldest of American petroleum fields, and although by no means a large one, is still an important one, because of the high paraffin yield in its lubricating oil fraction.

The Lima-Indiana field embraces the deposits of northwestern Ohio, Michigan and northern and central Illinois. Predominantly these oils are of a lower grade than the Appalachian crudes, especially as regards their high sulphur content which requires special refining methods for its removal. Colors range from green to brown.

The Illinois field extends from central Illinois to southern Indiana. The crude oils in this area are widely dispersed and exhibit a great variation as to quality, ranging from heavy high-sulphur asphaltic-base oils in the northern part to lighter and better material in the south.

The Mid-Continent is the largest American oil-producing area and includes all of Kansas and Oklahoma and northern, western and central Texas, as well as northern Louisiana and southern Arkansas. The crudes vary widely in passing from one extreme of this territory to another; some Louisiana wells yield a thick black oil of 21 degrees A.P.I. while the famous "gasoline well" near Cushing, Oklahoma, gives an almost colorless liquid of 55 A.P.I.

The Gulf Coastal region is constituted of those fields in Texas and Louisiana which border upon the Gulf of Mexico. This area has been and continues to be an enormous producer, with crudes tending toward asphaltic base and high sulphur.

The mountain states of Colorado, Wyoming and Montana, with small areas in Utah and New Mexico make up the Rocky Mountain district. Two types of oil are produced, paraffin- and asphalt-base oils. Most of these oils are dark and heavy, some running as low as 11 A.P.I.; a few are remarkably light in

Source	A.P.I. Deg. aver.	Color	Base	Type of hydro- carbon	Ultimate analysis, per cent				Soybolt universal viscosity at 100° F. X	Geologic period	Rock
					Carbon	Hydro- gen	Sulphur	Oxygen & nitrogen			
Appalachian	28 43	53	Amber to black	Paraffin	84.9	13.7	0.1	1.4	41 48	Devonian and carboniferous	Sandstone
Lima-Indiana	30	Green to brown	Paraffin	Paraffin					35.8	Ordovician	Limestone
Illinois	27 33	37	Brown to black	Par. and mixed			0.24		32.5	Carboniferous	Sandstone
Michigan	44			Paraffin						Silurian	Sandstone
Oklahoma	34 55	40	Green	Paraffin	85.7	13.1	0.4	0.3	34 45	Carboniferous	Sandstone
Kansas	35				84.2	13.0	1.9	0.45	54 390	Carboniferous	Sandstone
North Texas	32 43	35	Green	Paraffin			0.65		40 105	Carboniferous	Sandstone
Texas and Louisiana Gulf	15 30	22	Brown to black	Asphalt Naph- thene	84.6	10.9	1.63	2.9	38 540	Cenozoic and cretaceous	Dolomite and sandstone
North Louisiana	21		Black				0.4		85 265	Cenozoic and cretaceous	Sandstone
Wyoming	12 44	32	Green	Par. and mixed			2.0		33 1700	Cenozoic and cretaceous	Sandstone
Colorado	40		Brown green	Paraffin			0.1		250 36 50	Cretaceous	Sandstone and shale
California	10 54	21	Yellow to black	Asphalt Naph- thene			0.56	<32 240 50	Cenozoic	Sandstone and conglomerate	

FIG. 2.—Characteristics of crude petroleums.

color with gravities as high as 50 A.P.I. Wyoming is the main producer for this region.

The California district is usually considered as having two distinct series of pools—the San Joaquin or Valley fields, and the Coastal fields. The crudes are usually asphalt base, with low sulphur content, ranging in color from black to yellow.

REFINING OF PETROLEUM

Since petroleum is a complex mixture of hydrocarbons, together with compounds of oxygen, nitrogen and sulphur, its refining is a process of separating the hydrocarbons into groups of related compounds and of removing the impurities. Although refining operations are diverse and varied and although refining methods vary widely, as do also the types of apparatus employed, the most commonly used principle of refining is that which separates the petroleum into groups of hydrocarbons, or fractions, according to the range of boiling points.

It was pointed out under the discussion of hydrocarbons that the boiling points of successive members of any series showed gradually increasing temperatures, and it was also explained under the definition of a liquid that every substance when in the liquid phase has a definite boiling point, but that the boiling point is increased when the one liquid is admixed with another or with a dissolved substance. Hence a complex mixture like petroleum will behave as follows when subjected to temperature high enough to induce boiling.

As the temperature of the liquid is raised, its vapor pressure increases, and as soon as that vapor pressure equals that of the environment above the liquid, boiling commences. Now it is only natural that the components of the liquid mixture which have the lowest boiling temperatures will first form vapor, or boil. If each component of the mixture were to retain its individual boiling point, then by closely regulating the heating it might be supposed that complete separation of each hydrocarbon might thus be effected. But the facts are that each individual boiling point is affected by the presence of the other compounds, so that such perfect separation cannot be accomplished. What actually happens, therefore, is that the vapors arising from the heated petroleum are the mixed vapors of hydrocarbons with closely related boiling points or vapor pres-

sures. So each component hydrocarbon of the petroleum thus gradually boils out of the petroleum, and as the lower boiling compounds leave the liquid, the temperature of the liquid rises.

The first petroleum refinery therefore consists of a still or vessel—in which a charge of petroleum is placed and under which a fire is applied—together with a system for conducting the vapors from the top of the still through a condenser and thence into a receiver. By changing the receiver at certain intervals, which are determined either by the temperature of the vapor or the specific gravity of the condensed distillate entering the receiver, a separation of the petroleum is made into any desired number of fractions.

Of course those impurities which are present in the petroleum also distill over in the fractions corresponding to their boiling characteristics. The second part of the refining operation is therefore the removal of undesirable impurities. These refining processes are quite varied in their nature and involve subjecting the fraction to some physical or chemical treatment. Among the physical operations are filtration, absorption and dissolving by a solvent. In chemical processes the fraction is treated with some chemical, frequently sulphuric acid, which combines with the foreign material in the hydrocarbons. In the latter case, surplus acid can be removed by treatment with an alkali, such as caustic soda, followed by a final washing with water to clear out the alkali.

The batch or intermittent type of still was, with numerous improvements, used for many years in the petroleum industry, and still is used, although within the past ten years the continuous or pipe still has been rapidly displacing it. It will be instructive however to study in closer detail the products resulting from batch-still operation. First, however, some important modifications of the method must be noted.

After the lighter or low-boiling components have been distilled out of the charge, the temperature rises to send over the medium fractions. Further heating or continuance of the process will send the temperature to so high a point that chemical changes begin to occur in the remainder of the charge and the compounds originally present in the crude begin to decompose into smaller molecules. To avoid this, other means are resorted to that will bring about vaporization besides heat alone. Of these, there

are two: (1) introducing steam into the liquid and (2) diminishing the pressure within the vessel to create a partial vacuum; or both processes may be used. Under these conditions, it is possible to vaporize a liquid at a temperature lower than that at which it would otherwise occur; for in the case of steam introduction, the vapor pressure of the steam adds to that of the hydrocarbons to overcome the pressure of the environment above

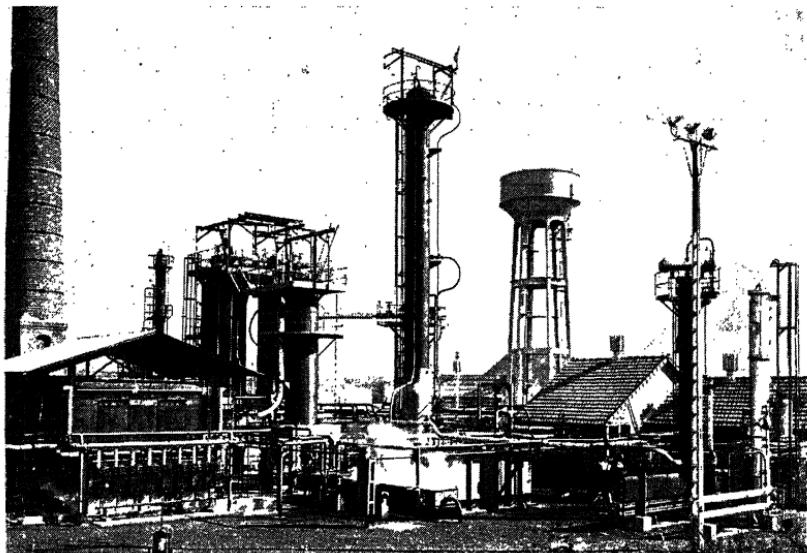


FIG. 3.—Modern Dubbs cracking unit. (*Courtesy of Universal Oil Products Company, Chicago.*)

the liquid, and in the case of creating a vacuum, the pressure of the environment is reduced so that the vapor pressure of the liquid becomes equal to that of the environment at a lower temperature, permitting of boiling below the atmospheric boiling point. When both steam and vacuum are employed, of course, the benefits are additive.

The first fraction collected from a batch distillation is called the naphtha and gasoline fraction. For many years this mixture was termed benzine, and as such had industrial applications as a solvent in cleaning, dyeing and painting. The term benzine can easily be confused, however, with benzene, the aromatic compound C_6H_6 , so that the term has gradually been abandoned in favor of the name naphtha. The naphtha fraction is cut at a

vapor temperature of about 400°F. The naphtha fraction is in turn subjected to additional refining, either distillation or chemical treatment or both, until in general a division has been made into motor gasoline and cleaners' or solvent naphtha.

The second cut is made at a vapor temperature of about 525°F.; this fraction when again purified yields the water-white distillate called kerosene.

Immediately following the kerosene, another cut is taken which is variously called solar oil, furnace oil or distillate oil. Formerly this was not usually subjected to additional treatment, but in modern practice it is cut further. From this fraction are obtained distillate fuel oils used for both heating fuel and internal-combustion engines of the tractor type.

During the evaporation and collection of the next fraction, if neither steam nor vacuum be used, destructive distillation occurs, and the products would depend upon the rate of heating. This gas-oil fraction, as it is called, will run an average of about 23 to 26 A.P.I.; if the temperature is allowed to rise high enough to accomplish the boiling unaided, the material will contain cracked products. If steam or vacuum be employed, then decomposition is avoided and uncracked hydrocarbons are recovered. This fraction is very important in formal cracking processes, as will be brought out below.

Wax distillate comes over from the still after the gas oil. It furnishes the stock from which lubricating oils and wax are made. The wax is removed by operations of cooling, pressing, melting and filtering. The wax-free oil is called pressed distillate and is blended in varying proportions with bright stock to make various grades of lubricating oil. The wax which is separated is refined into the paraffin wax of commerce.

The residue remaining in the still, if the distillation has not been carried too far, is heavy fuel oil or tar, depending on how much fluid has been driven off and also on the nature of the crude. Incidentally, this residue is the basis for the classification of crude oils. Should the remaining material from the distillation be waxlike, the crude is termed paraffin base; should it be asphalt the crude is termed asphalt base; sometimes the two occur together and the crude is then designated as a mixed base crude.

Should the distillation itself be carried on until all liquid has been expelled from the still, the ultimate residue is petroleum

coke. This is greatly valued as a domestic fuel, as it can readily be burned in small stoves and furnaces. Because of the oil with which it remains impregnated, this coke has a very high calorific value per pound.

The description given above for petroleum refining is not complete in that it lacks three elements—elements which are the achievements of modern technology—cracking, fractionation column, and pipe still. Each is now described below—in inverse order.

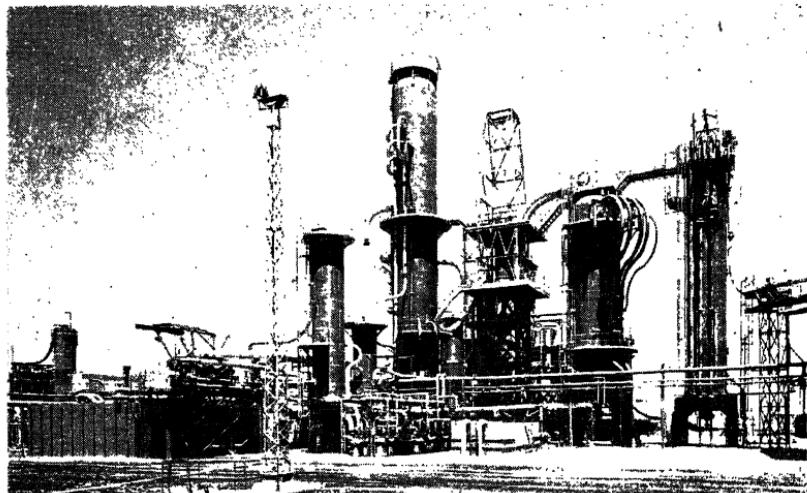
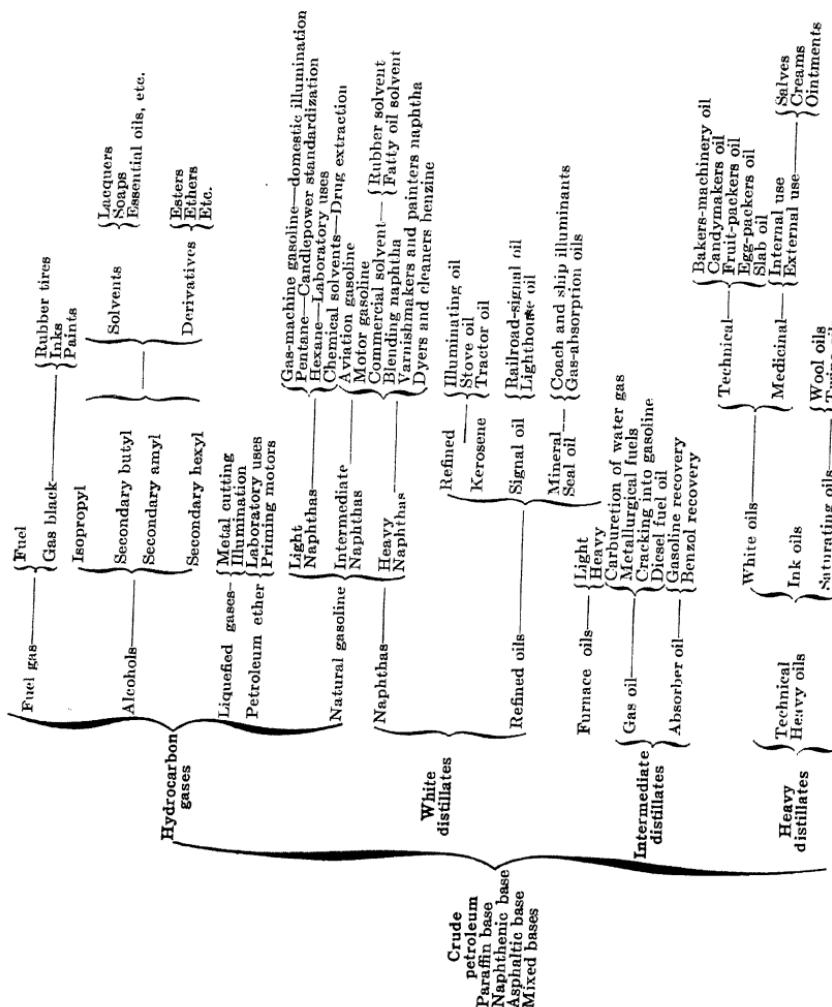


FIG. 4.—A modern refinery; Dubbs crude topping and selective cracking unit.
(Courtesy of Universal Oil Products Company, Chicago.)

First, it should be mentioned that the batch process which has been discussed is not necessarily limited to a single still for the entire distillation. Certain fractions may be taken off the crude in one still, and the remaining liquid then pumped into one or more successive stills for the evaporation of the other fractions. This tends to make the process a continuous one, and obviates the necessity for cooling down and cleaning out a still after running a single batch or charge. The modern pipe still is in reality a series of such successive stills combined. By means of it, petroleum may be carried through a series of exposures to gradually increasing temperatures so that the various components may be extracted without the necessity of transferring the charge from one still to another.



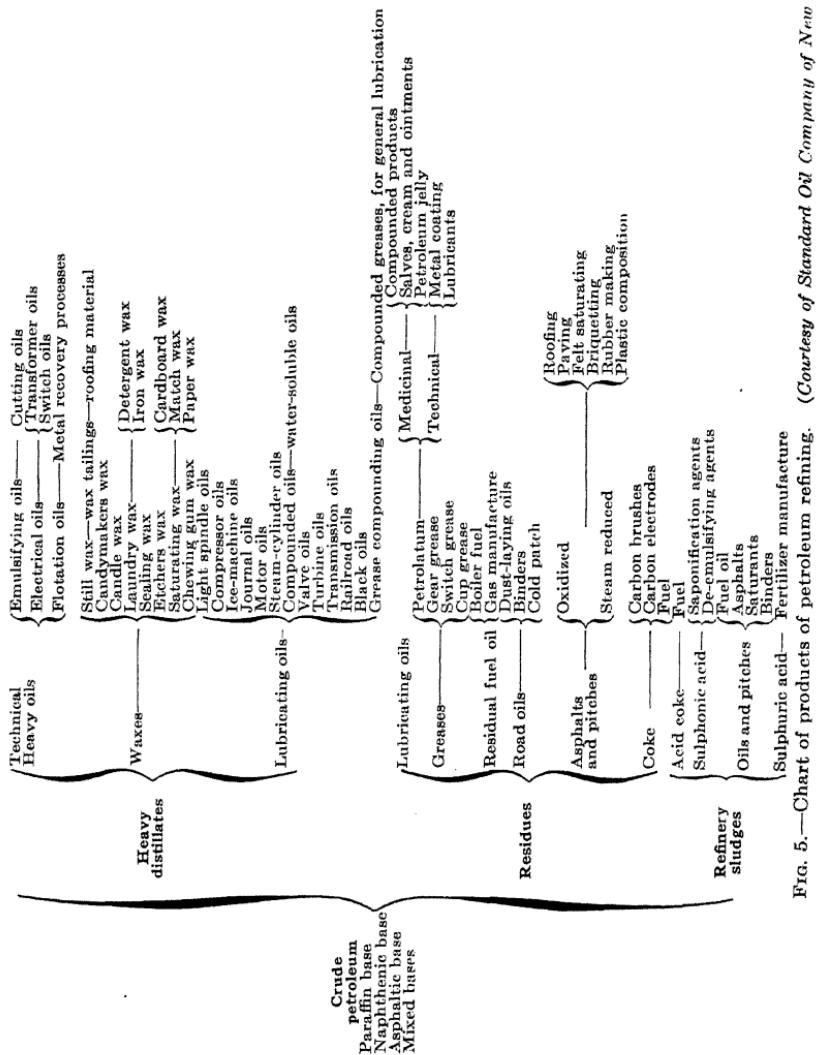


FIG. 5.—Chart of products of petroleum refining. (Courtesy of Standard Oil Company of New Jersey.)

As a matter of fact, the pipe still was tried out many years ago, when the shortcomings of the batch still were first realized. But it could then be applied only to topping crudes, *i.e.*, removing the low-boiling gasoline. Complete distillations could not be carried through for want of the device which now makes it possible—the fractionation column.

It has already been mentioned that the process of distilling does not permit of making a clean-cut separation of individual compounds, because of the overlapping of boiling points and the influence of one substance upon the boiling point of another. Another factor which makes difficult the separation of substances is entrainment, or the bodily carrying up of small globules of unevaporated liquid with the true vapors. If now the vapors and entrained globules of liquid are immediately taken into the condenser and reduced to the liquid state again, the result is a mixture having a considerable range of boiling points. But if the vapor is permitted to exist for a period of time under suitable conditions, the entrained liquid will separate and fall out, and the vapors of the higher boiling constituents will condense out and return to the liquid phase. A fractionating column is an apparatus which makes this possible.

In practice, fractionating columns are tall cylindrical extensions of the still proper, either batch or pipe type, rising vertically. Stacked up in the column are a great many baffles—and in form a great many types of baffles have been tried. Regardless of its particular construction, the function of the baffle is to impede the path of the vapors so that entrained liquid globules will be physically thrown out and fall back into the still. As the vapors rise, the higher boiling components recondense and also fall back into the still. When the vapors have reached the top of the column, therefore, a considerable purification has been accomplished, and the resultant distillate condensed in the condenser has a much closer boiling range than would otherwise be possible.

But the crowning achievement of petroleum refining was the invention and development of the art of cracking. It has increased the average yield of gasoline from about 20 to about 40 per cent of the crude.

Briefly, cracking consists in subjecting the gas-oil fraction of an original distillation to heat and pressure which bring about a decomposition of the heavy hydrocarbons into lighter compounds.

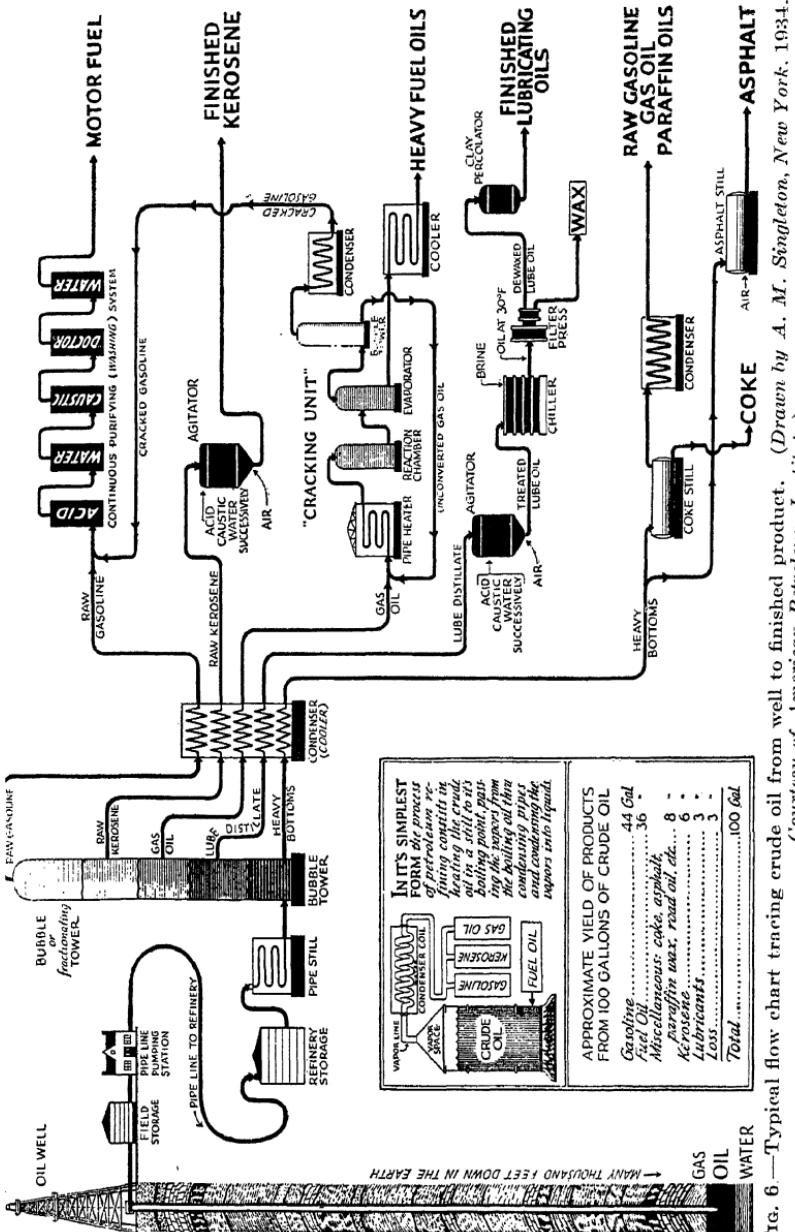


Fig. 6.—Typical flow chart tracing crude oil from well to finished product. (*Drawn by A. M. Singleton, New York, 1934.*
Courtesy of American Petroleum Institute.)

This cracked gas oil is then introduced into a still to be fractionated. As high a yield of gasoline as 80 per cent can be secured from cracked gas oil.

Insofar as fuel oils are concerned, recent practice has been to make the first three grades, Nos. 1 to 3, overhead products, *i.e.*, these are materials which have passed out of the still in the vapor phase and then recondensed. The last three, Nos. 4 to 6, may be residuum.

This brief discussion of the fascinating subject of petroleum refining has hardly scratched the surface of the actual technology involved. It is earnestly hoped that the reader seeking broader knowledge will consult the references below.

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CHAPTER II

FUEL OIL

Fuel oil is a mixture of hydrocarbons whose volatility is too low to permit its use as a carbureted fuel in an internal-combustion engine, except the Diesel engine. The type of hydrocarbons constituting fuel oil may vary considerably, and the chemical and physical properties of the fuel vary accordingly. To make an intelligent study of fuel oil, therefore, it is necessary to investigate the effects of the type of hydrocarbons upon the nature of fuel oil.

The most frequently occurring series is the paraffin group. It will be recalled that the first four members of this series are gases; methane, ethane, propane, butane. The boiling point of butane, however, is 31°F., so that at lower temperatures it is a liquid. The fifth member, pentane, boils at 96.8°F. and so is a very volatile liquid. The succeeding members are all liquids up to at least pentadecane, $C_{15}H_{32}$, which melts at 50°F., and beyond that, higher members become more and more waxlike at ordinary temperatures. When the series reaches pentatriacontane, $C_{35}H_{72}$, the melting points are around 167°F. In a similar way, with the olefins, ethylene, propylene and butylene are gases, and the higher members follow the paraffins fairly closely in the transition to liquids, and to solids. In general therefore, it is customary to consider the chain hydrocarbons as belonging to the gaseous fuels up to 4 carbon atoms in the chain, in the gasoline group up to 11 carbon atoms in the chain, and in the lubricating- and fuel-oil classification up to about 25 carbon-atom chains. Thereafter the compounds generally must be considered as asphalt or tar.

The progressive transitions of physical characteristics are clearly seen from the tabulation on p. 32 of data on commercial liquids.

Because of this general tendency of physical characteristics to vary according to the progressive increase of the carbon chain, there has arisen a practice in the industry of judging fuel

oils entirely from a few relatively simple tests. During the early years of the petroleum and oil-burner industries, this may have served a purpose, but with the advent of modern oil-burning equipment and with the demands of present-day competitive fuel markets, these tests no longer serve to convey an adequate description of a fuel oil. A much closer scrutiny is necessary. In order that closer observations of fuel oils may be made, and

Classification	Specific gravity	A.P.I. degrees	Initial boiling point,	End point,
Gasoline.....	0.7238	64	100	435
Naphtha.....	0.7796	50	280	440
Kerosene.....	0.8155	42	305	590
Distillate.....	0.8348	38	420*	600
Gas oil.....	0.8498	35	420*	650
Residual fuel oil.	0.9340	20		

* Ten per cent distilled over.

more critical judgment be passed upon them, and in order that the behavior characteristics of a fuel oil may be foretold, it is essential that the chemistry and physics of fuel oils be investigated and that the significance of chemical composition and physical properties be interpreted from the viewpoint of applicability.

CHEMICAL PROPERTIES OF FUEL OIL

STABILITY

It has been established that the paraffins are a chemically inert group of compounds. By that is meant that these substances do not readily undergo or enter into chemical reactions with other substances. Another way of expressing the same property is to say that the paraffins possess chemical stability. As unsaturatedness occurs, chemical stability is impaired and the possibility of chemical activity begins. Since there are relative degrees of unsaturatedness, there are correspondingly relative degrees of chemical stability. Ranging down from the paraffins, in order of stability, there are naphthenes, aromatics, olefins and acetylenes. It can be said that the paraffins represent the best conditions of chemical stability and the acetylenes the worst.

The significance of stability can be understood when certain facts are considered. Petroleum products are passed through a large variety of processes and are handled and transported and stored a great many times before arriving at the point of ultimate consumption, and even there physical handling occurs. During this time, the chemical compounds comprising the petroleum product may be subjected to high temperatures and contact with other substances ranging from atmospheric oxygen to strong chemical reagents. Being no stronger than its weakest link, the carbon chain cannot resist more than can the double bonds it contains. Hence when unsaturated compounds with marked chemical activity are acted upon by, or subjected to, chemical influences, chemical change may occur at those points in atomic structure where double bonds may occur. Should an atom of oxygen be introduced into the hydrocarbon molecule, by reason of varied and perhaps complex chemical changes, the oxygen atom creates a marked difference in the chemical nature of the substance. It has already been shown that a single atom of oxygen, however introduced into the molecule, will convert methane into wood alcohol; in a similar way an oxygen atom entering, at a point where a double bond exists in an olefin, into a chain of about ten carbon atoms would convert a hydrocarbon into an asphaltene, which is insoluble in some petroleum products, as for instance naphtha. Or again, under the influence of heat, two or more olefins may unite at their respective double bonds, forming a compound of great chain length and complexity, and thereby change a liquid into a solid. Hence the foremost consideration of chemical properties is the one of stability.

In the lighter liquid fuels, as gasoline and kerosene, the formation of oxygen addition products as suggested in the last paragraph is termed gum formation. These gums can be much more readily detected in a clear liquid, like gasoline, than in a black-colored fuel oil. There is a possibility of such oxygen compounds being formed within the oil after leaving the refineries. If the fuel oil is rather heavy, it is quite possible that asphaltenes will remain in solution, but should this heavy material be subsequently blended with a lighter fuel oil, precipitation may occur by reason of the insolubility of the asphaltene in the lighter fuel oil.

Chemical stability is thus an indication of the likelihood of change occurring in the physical characteristics of the fuel prior

to the moment that combustion occurs. Stability has little effect upon combustion itself; most hydrocarbons of equivalent carbon-chain length burn with equal ease regardless of chemical composition. It will be seen later that flash and fire points and viscosity are better indications of fuel oil combustion properties.

Sulphur burns to form sulphur dioxide, SO_2 , which is a gas that is readily soluble in water. Upon dissolving in water it forms an acid, sulphurous, H_2SO_3 . Sulphurous acid must not be confused with sulphuric acid, H_2SO_4 . This latter acid, sulphuric, is formed when sulphur trioxide, SO_3 , is dissolved in water. But the trioxide cannot easily be prepared and does not form at all when sulphur burns under ordinary conditions. But although not so powerful or corrosive an acid as sulphuric, sulphurous acid has marked and pronounced tendencies toward corrosion and will react with many metallic materials. Ordinarily, the sulphur dioxide formed in flue gases passes out of the chimney without causing any appreciable damage, but under certain conditions, it can do extensive damage. Should the products of combustion be cooled below the condensation point of the water which is always formed when hydrocarbons burn, this water will dissolve the sulphur dioxide and form the corrosive sulphurous acid. This will in turn act upon metal breechings and stacks by dissolving the metal at the surface of contact. Again, in metallurgical and ceramic furnaces, sulphur dioxide will be absorbed by molten metal or incandescent clay products, to the detriment of the product. In most metallurgical work, sulphur content of the product must be held within very close limits; high sulphur fuels will interfere with this. In other industrial processes, such as baking, sulphur in fuel oil is equally objectionable.

Sulphur may be present in various forms. The compounds known as mercaptans have been mentioned in Chap. I. They will reveal their presence by pronounced odor. Hydrogen sulphide, H_2S , is a common form of sulphur compound encountered in petroleum; sulphonic acids are sometimes present, particularly in the case of aromatics. Whatever may be the original form of the sulphur, however, the ultimate product of combustion is always sulphur dioxide.

Nitrogen, if present in petroleum or one of its products, is usually in the form of complex organic compounds. It does

not enter into chemical reactions of interest to fuel chemistry or fuel utilization. Oxygen compounds, such as originally may be in the fuel oil, may be similar in nature to asphaltenes which form by oxidation of olefins. Like nitrogen, oxygen is of little interest in the study of fuel chemistry. Other chemical substances than those already mentioned are usually mineral in nature and are termed sediment. Mineral matter is incom-bustible and forms an ash if introduced into a firebox. This ash may settle on the flues or heating surfaces of a boiler, but more generally is carried up the stack.

PHYSICAL PROPERTIES OF FUEL OIL

SPECIFIC GRAVITY

One of the most important physical properties of fuel oil is its specific gravity. In fact for many years the petroleum and oil-burner industries based the ultimate classification of fuel oils entirely on specific gravity, or, as it is loosely called, gravity. There appears to be some confusion as to the proper meaning and significance of the terms applied to the properties pertaining to weight, such as specific gravity, density and mass, so that a brief review of the fundamental physics will no doubt be helpful. The following discussion is not an exact scientific treatment but it should clear up any ambiguities on the subject.

If a physical body be placed upon a platform scale, the attraction of gravity which is exerted upon all objects causes the body to register a certain magnitude upon the scale. This is called its weight. Weight is created by the attraction of gravity, for it is manifest that, if the force of gravity did not exist, an object would not register a weight upon a scale. Weight is therefore a force acting downward in the vertical direction. The common unit of weight in engineering work is the pound; in scientific fields the weight unit is the gram.

All objects possess a property akin to weight, yet radically different from it. Imagine a body on a sheet of frictionless ice, at rest. In order to place this body in motion a horizontal force must be applied to it, and the greater the force, the greater the motion imparted to the body. This horizontal force cannot be acting against gravity since a vertical force such as gravity cannot have a horizontal effect. The horizontal force is expended

in acting upon the *mass* of the body, and the mass of the body can be measured and is technically defined in terms of the acceleration imparted to it by a unit horizontal force. In most fields of engineering, notably hydraulics and mechanics, the distinction between weight and mass is important.

Because weight is created by the attraction of gravity, the term gravity is often used synonymously for weight, but as such it is a loose and indefinite term. A more concise expression is the term specific gravity, and yet specific gravity expresses or implies weight only indirectly or by inference. For an absolute expression of the concept of weight which cannot be confused with other terms or concepts we should use the word density. The application of the expressions density and specific gravity are distinguished in Chap. XII, which deals with the hydraulics of oil flow.

Density means the number of units of weight in a unit volume of a substance or body. Since in engineering the unit of weight is the pound and the unit of volume is the cubic foot, it follows that, in engineering, expression of density is in pounds per cubic foot. Thus the density of pure water is 62.4; of wrought iron, 490; of aluminum, 168; of firebrick, 107. In metric measure, universally used in scientific procedure, the respective units are the gram, cubic centimeter and grams per cubic centimeter. Metric densities for the materials just mentioned are: water, 1; wrought iron, 7.85; aluminum, 2.69; firebrick, 1.71.

The specific gravity of a body or substance is the ratio of its density to that of some other substance which is considered the standard. For solids and liquids, and sometimes for gases, the standard substance is pure water; thus the specific gravity of wrought iron is $490/62.4$ or 7.85, of aluminum $168/62.4$ or 2.69, of firebrick $107/62.4$ or 1.71. It will be noted that specific gravity being a ratio has no units, and numerically is identical with density in metric units.

Most liquids are lighter than water, a fact which is immediately indicated by specific gravity values less than one; alcohol, 0.806; hexane, 0.655; octane, 0.699; benzene, 0.8724. In general, all petroleum crudes and derivatives are lighter than water, except for very viscous residuums and coke.

The specific gravities of gases when referred to water are, of course, very small fractions (specific gravity of ethane, for

instance, is 0.0013), so that when dealing with gases it is more common to refer gravities to air as the standard substance. Referred to air, the specific gravity of ethane is 1.03. The specific gravity of any gas may be determined by dividing its molecular weight by the molecular weight of air which is 28.85.

Because specific gravity is so useful an index of certain properties of fuel oil and other petroleum derivatives and because the decimal fraction which expresses the specific gravity of most liquids is rather cumbersome for frequent reference, a purely arbitrary scale for expressing this value, originally devised by the French chemist Antoine Baumé early in the nineteenth century, was adopted by the industry. This is known as the Baumé scale. It was first devised for work on hydrometers and, of course, is still extensively so used. The zero mark on such a hydrometer (for liquids lighter than water) is the point to which the instrument sinks in a 10 per cent salt solution, while the 10° mark is the point to which it sinks in pure water. The intermediate space on the hydrometer stem is then divided into 10 equal parts, and each designated as 1° Baumé. The scale is then extended beyond the 10° mark with the same division for successive degrees over 10. But the progressive increase of specific gravity of salt solutions, or most other physically varying sequences of liquids and their properties is not in such simple proportion, so that there is no simple relationship between actual specific gravity and degrees Baumé. The closest approximation that has been made gives a relationship of

$$S = \frac{140}{130 + B}$$

where S is specific gravity and B is degrees Baumé.

In a similar way the hydrometer scale for liquids heavier than water is established with the zero point at pure water and the 15° point at 15 per cent salt solution. The relationship for this scale is approximately

$$S = \frac{145}{145 - B}$$

In recent years the American Petroleum Institute has promulgated a slightly different scale in which the modulus, as the number 140 is called, is changed to 141.5, and the number

130 becomes 131.5. The original Baumé scale has largely been supplanted in the American petroleum industry by this newer scale for expressing specific gravity in terms of arbitrary degrees. Hence

$$\text{Degrees A.P.I.} = \frac{141.5}{S} - 131.5$$

In order to avoid confusion of terms relating to the relative weights of liquids, it is best to restrict the use of each term. Density should be used only in its technical sense of implying weight per unit of volume, pounds per cubic foot. Specific gravity should be used only when it is intended to refer to the actual ratio of the weight of a cubic foot of liquid to 62.4 (the weight of a cubic foot of water). Specific gravity is invariably expressed as a decimal fraction. The use of the word gravity, or specific gravity, should be avoided for conveying the thought of degrees Baumé, or degrees on the A.P.I. scale. Again, degrees Baumé should not be mistakenly stated when degrees A.P.I. (American Petroleum Institute) is the correct expression of the scale used. In general, since the original Baumé scale is rapidly passing out of use, the best method of stating or writing the desired expression for the indicated reading on the A.P.I. scale is "degrees A.P.I."

Thus, a fuel oil having a density of 55.755 lb. per cu. ft. has a specific gravity of 0.8927. On an hydrometer it will show a reading of 27 degrees A.P.I. In less formal nomenclature, especially in the loose diction of everyday routine, it is probable that the word gravity will persist, and a fuel oil will be spoken of as having a gravity of 27. But it should be clearly understood that the scale almost universally used in America is the A.P.I. scale, and not the old Baumé scale. At least, the word Baumé should be dropped entirely from usage.

THERMAL EXPANSION

All liquids, in common with solids and gases, expand and contract with temperature changes. Hence a statement of density or specific gravity is concise only if the temperature of the liquid is given, as well as the observed specific gravity or degrees A.P.I. In the petroleum industry, it is customary to use 60°F. as the standard temperature, and in commercial

transactions, all quantities are referred back to this datum. Tables are given in the Appendix showing corrections for temperature for observed degrees A.P.I. The following formula gives a method for correcting any volume back to a standard temperature or to translate a volume at one temperature into volume at any other temperature.

$$V_2 = V_1 + aV_1(t_2 - t_1)$$

in which V_2 is volume at temperature t_2 , V_1 is volume at temperature t_1 and a is the coefficient of cubical expansion (see Appendix for values).

VISCOSITY

Insofar as the commercial and industrial grades of fuel oil are concerned, viscosity is the most important single physical attribute. In fact, of the six grades of fuel oil set up by customary present classifications, only Grades 1 and 2 are independent of viscosity. It might be expected, therefore, that of so significant a property there would have been extensive observations and investigations made. Such is actually the case; and unless careful study is made of the somewhat extensive data and literature on the subject, it might appear to be a prolix and complex matter. But if a few generalizations are first laid down as a guide, most of the facts can be aligned along a fairly direct course of relationship.

Viscosity is variously defined as the internal friction of a liquid; as the measure of its resistance to flow; as the resistance experienced by one portion of a liquid in moving over another portion; as the rate of yielding of a liquid to a force tending to change its shape. Scientifically, viscosity has been measured by observing the force required to move the upper of two parallel plates of unit area, at unit distance apart, at unit velocity, with the space between the plates filled with the liquid under test. The force so measured is called the coefficient of viscosity. When the unit area of the plate is 1 sq. cm., the unit distance between plates 1 cm., the unit velocity 1 cm. per sec., then the unit coefficient of viscosity is called a poise.

Actually to measure viscosities of liquids by a method conforming to the above scientific definition would not be practicable. For commercial purposes there have been developed various

forms of instruments known as viscosimeters. These instruments do not measure viscosity directly—they measure rather the effect of viscosity upon the time of efflux of a liquid through a capillary tube. It was shown by the French hydraulician Poiseuille, about 1843, that if a volume of liquid V flows through a capillary tube of radius r and length l in time t under a pressure P , where the coefficient of viscosity is Z , then

$$Z = \frac{\pi r^4}{8Vl} t$$

Instead of calibrating viscosimeters under this formula and measuring viscosities in poises, it is customary in industry to employ instruments of fixed outlets or capillaries and then merely to determine the time in seconds for a standard volume to flow through, at a definite temperature. In the United States the two most generally used instruments are the Saybolt universal and the Saybolt furol; in England the Redwood is largely used; and in Germany the Engler is the most generally used. Viscosities so determined are expressed in terms of time of efflux in seconds: as 147 sec. Saybolt universal at 100°F., or 62 sec. Saybolt furol at 122°F. Since viscosities of fuel oil, in the United States, are usually expressed as seconds Saybolt universal at 100°F., except for very heavy grades of No. 6 which are usually given in seconds Saybolt furol at 122°F., it is convenient to remember that the former is approximately 20 times the numerical value of the latter. In the case of the above example, therefore, the 62 sec. Saybolt furol at 122°F. would correspond, approximately, to 1240 sec. Saybolt universal at 100°F.

If the viscosity of a liquid in poises be divided by the liquid's specific gravity at the same temperature, the resultant expression is called kinematic viscosity, *i.e.*, $k = Z/\text{sp. gr.}$ The kinematic viscosity in turn can be expressed in terms of Saybolt universal seconds, by the relationship

$$k = 0.0026t - \frac{1.95}{t}, \text{ when } t \text{ is 100 or less}$$

and

$$k = 0.0020t - \frac{1.35}{t}, \text{ when } t \text{ is over 100}$$

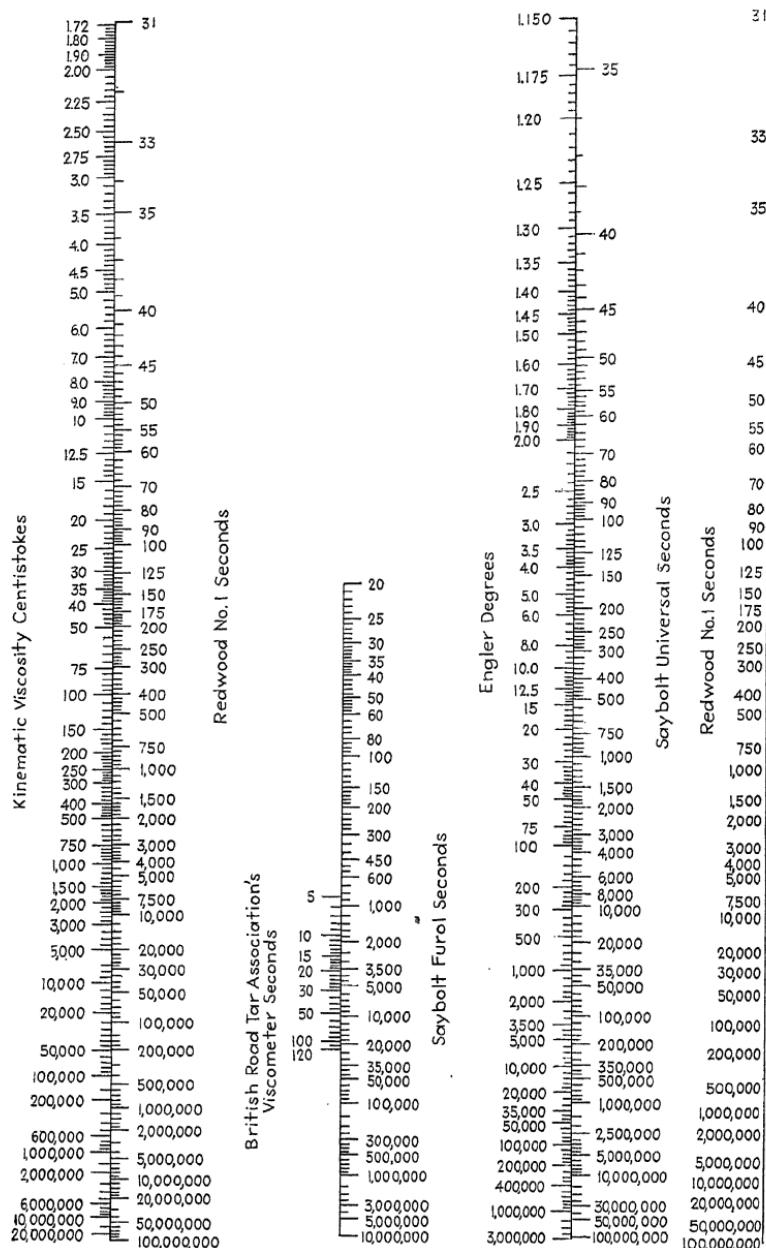


FIG. 7.—Conversion chart for various viscosity scales. (Copyright Precision Scientific Company.)

t being time in seconds Saybolt universal. The unit of kinematic viscosity is called the stoke.

Measured in poises, the viscosity of pure water at various temperatures is: at 32°F., 0.0179; at 50°F., 0.0131; at 68.4°F., 0.0100; at 100°F., 0.0069; at 160°F., 0.0040. Because the value at 68.4 is just 0.01, this value is often used as a standard refer-

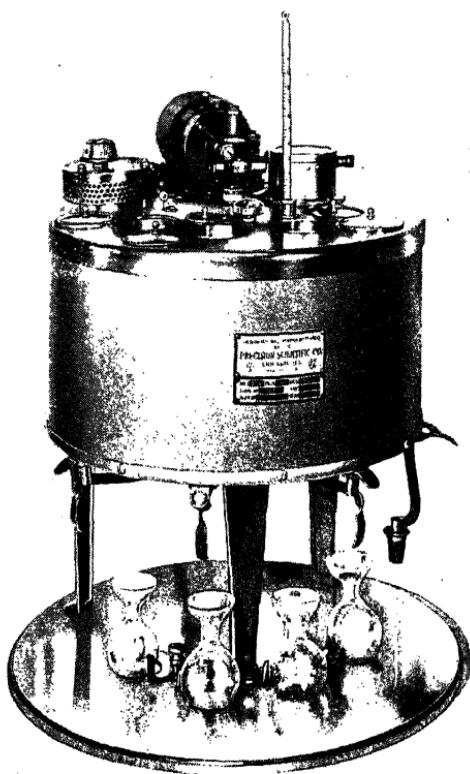


FIG. 8.—Saybolt viscosimeter. (*Courtesy of Precision Scientific Company.*)

ence, and is termed a centipoise. The corresponding value of kinematic viscosity is termed a centistoke.

The theoretical considerations of viscosity are chiefly of value insofar as they affect the flow of fluids in conduits and pipes as shown in Chap. XII. The practical value of viscosity lies in judging fuel oils and classifying them according to the recognized grades. The accompanying chart (Fig. 7) shows conversion

values between the various viscosity scales, both European and American.

It has been established that the variation of the viscosity of petroleum derivatives is in logarithmic proportion to temperatures. The American Society for Testing Materials has prepared a chart which enables the laying of a straight line to show the viscosity-temperature relationship for a petroleum material if two sets of values are known (see Fig. 191). The same authority gives a method for graphically determining the viscosity of a mixture from the percentage composition and viscosity of the two materials admixed.¹

SURFACE TENSION

Surface tension has not received the attention or study which is probably warranted in view of its potential bearing on the subject of fuel atomization. A brief review of the general principles of physics which deal with surface tension will show that the present knowledge on this subject and the conclusions to be drawn therefrom indicate that many of the phenomena of oil burning should be explainable through interpretation of surface tension. Much investigation and research are here required, however.

By surface tension is meant the peculiar manifestation of molecular attraction which exists at the surface of a liquid—commonly known as skin effects. Thus most liquids will wet the sides of a vessel containing them and will crawl up the sides of the vessel to form a meniscus. Similarly, those liquids which do not wet the walls of the containing vessel are depressed at the perimeter of contact, to form a meniscus concave downward. Mercury is a familiar example of this. The strength of the meniscus, in the case of liquids which wet retaining walls, will cause the liquid to rise in a small vertical tube (called a capillary tube) above the level of the main body of the liquid in which the capillary is inserted. In the case of mercury, the meniscus will cause the liquid to be depressed when a capillary tube is inserted into the liquid body. Among the other manifestations of surface tension are: the tendency of liquids to assume a spherical form if free to do so, as when molten lead is permitted to fall freely

¹ A.S.T.M. standards on "Petroleum Products and Lubricants," Sept. 1934 (see Fig. 191).

through space until solidification occurs, or when small quantities of mercury are detached, or when small portions of one liquid are introduced into another immiscible with it; the regular formation of uniformly sized drops which occurs when a liquid is permitted to issue from a vertical tip of small diameter; the formation of bubbles by soap solution.

By means of extremely careful experiment and complex mathematics, physical chemists have established that there are definite relationships between the molecular weights of liquids and their surface-tension phenomena. Thus molecular weights have been computed from the size of drop formed by a pure liquid in issuing from a calibrated tip. Other physical properties interrelated with surface tension are coefficient of expansion and viscosity. It seems probable that someday there will be definitely established laws showing the interdependence of surface tension, viscosity and atomizing characteristics of a fuel oil, as well as the bearing of chemical composition on the matter.

PHYSICOCHEMICAL PROPERTIES OF FUEL OIL

Thus far the properties of fuel oils which have been discussed mainly have been either purely chemical or purely physical. A large and important group of properties may be termed physicochemical, as their nature and interpretation lie in the realms of both sciences—physics and chemistry. These are presented below, roughly in order of their relative importance.

DISTILLATION TEST

The distillation characteristics of a fuel oil are an indication of its chemical composition, since it is known that the boiling points of hydrocarbons increase with their molecular weights. Much valuable information can therefore be secured from studying the distillation curves of fuel oil, but this method of investigation is feasible only on the lighter grades, as hydrocarbons begin to decompose at elevated temperatures. Hence this method is restricted to materials which can be entirely or almost entirely distilled below this temperature.

The customary commercial method of conducting a distillation test is to observe the temperatures of occurrence of: the initial boiling point, *i.e.*, the point at which the first drop of distillate or recondensed fuel oil falls from the end of the condenser tube;

the successive points at which all multiples of 10 per cent have been collected in the graduated receiver; the end point, or temperature at which the last particle of liquid remaining in the distillation flask is vaporized and nothing remains but dry residue. The total amount of distillate collected in the receiver, representing the recovery percentage, is also observed and recorded.

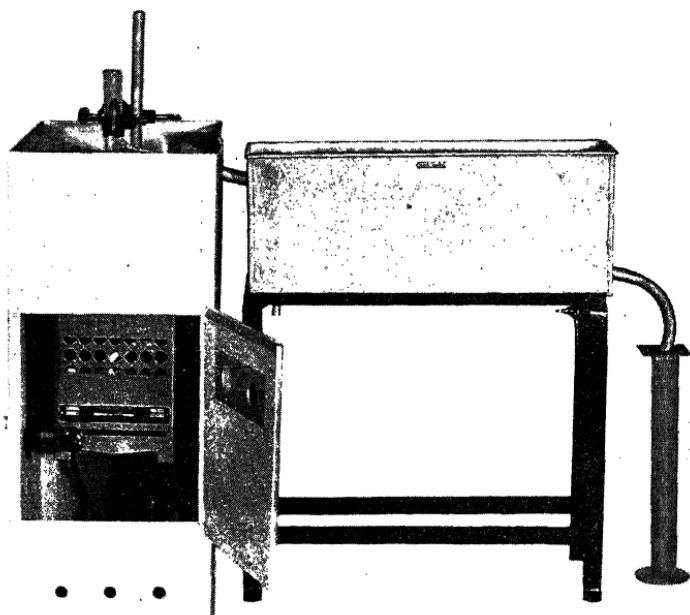


FIG. 9.—Distillation apparatus. (*Courtesy of Precision Scientific Company.*)

When the various multiples of 10 per cent recovery temperatures are plotted on cross-section paper, the result should be a smooth curve, indicating a uniform rise of temperature with progressive amounts of distillate recovered. Any abrupt change in slope of the curve indicates an abnormal composition of the fuel; a low recovery is an indication either of cracking having occurred during the distillation to yield noncondensable gases or of high nondistillate constituents in the fuel oil; an end point significantly higher than the 90 per cent temperature shows that some high-boiling material has been added to the original fuel.

For those burners in which the fuel oil is vaporized in order to accomplish combustion, the distillation range is of especial importance. The 10 per cent point tells whether ignition can be successfully and dependably attained, although the flash and ignition tests also have some bearing on this; the 90 per cent and end points are an indication of what may be expected as to completeness of combustion and freedom from carbon and soot. These points are elaborated upon in the discussion of burner types and atomizing principles.

It has been mentioned, in Chap. I, that pure liquids have definite melting points and boiling points, and that the temperatures at which these changes of state occur are influenced by the presence in the liquid of a dissolved impurity. In dealing with a mixture of liquids, it is somewhat inconvenient to consider any one of the pure substances as the main material and the others as impurities, whatever may be the percentages present. Instead, for the purpose of studying boiling phenomena, it is more convenient to deal with vapor pressures.

The molecules of a liquid are in constant motion; their average speed increases in direct proportion with the temperature. Certain molecules, in their undirected motion, leave the liquid entirely by moving out through the surface film. This is termed evaporation. Should the liquid be enclosed in a closed container, the molecules passing into the space above the liquid are unable to leave the confining space and so must either remain in this space or fall back into the liquid. At any given temperature, a state of equilibrium is soon reached, when the number of molecules leaving the liquid in unit time equals the number returning. The space above the liquid is then said to be saturated with the liquid's vapor. The *concentration of this vapor in the space above the liquid is dependent only upon the temperature and is independent of the nature of other substances present in this space.* Thus whether the space above the liquid be filled with air or any other gas, the number of molecules required to fill this space to the point of saturation for any particular liquid is the same for any given temperature. This is the famous law of Dalton on partial pressures and is stated as, *the total pressure of a mixture of gases is equal to the pressures which each would exert were it alone present in the volume occupied by the mixture.* The pressure which the vapor of a liquid exerts under these conditions is called its vapor

pressure. The vapor pressure of a liquid increases with temperature, and at the boiling point the vapor pressure becomes equal to atmospheric pressure. From this it can be seen clearly that a liquid will boil at a higher temperature if subjected to a pressure above atmospheric or at a lower temperature if the pressure is below atmospheric. It follows therefore that the temperature of steam at atmospheric pressure is 212°F., but that the steam is hotter if the boiler pressure is above zero gauge and cooler if the pressure is below zero gauge (a boiler condition which usually is described as a vacuum).

To return to the mixture of hydrocarbons composing a fuel oil, it can be seen now that each liquid present in the mixture is exerting its own vapor pressure, and each component exerts this vapor pressure independently of its percentage in the liquid. When the total of all the vapor pressure equals atmospheric, boiling commences. Hence a small amount of a low-boiling liquid present in a mixture with higher boiling substances will induce boiling at a lower temperature than otherwise. But as soon as this small percentage of low-boiling constituent has evaporated completely, the boiling characteristics become more normal to the remaining liquids.

FLASH AND FIRE POINTS

The flash point of an oil is the temperature at which evaporation occurs at a rate sufficiently rapid to permit a concentration in the space above of vapor which will ignite momentarily upon the application of a flame. Naturally, the exact temperature at which this occurs for any given oil depends upon the nature of the space above the liquid and upon the method of flame application. But for a fixed method of testing for flash, the presence of a low-boiling constituent, even though in very small concentration, will give a flash point which may be unreasonably low for the bulk of the material present. This is why a flash point cannot be accepted as complete evidence of an oil's volatility—just as the initial point of a distillation test may similarly be misleading. But the 10 per cent point of a distillation is much more accurately a reflection of an oil's characteristics.

If, after the flash point has been reached, the heating of an oil sample is continued, a temperature will be reached at which the evolution of vapors is sufficiently rapid to support a flame in

continuous burning when a light is applied. The most recent specifications for commercial grades of fuel oil do not include this fire point.

CLOUD, POUR, AND COLD TESTS

Another physicochemical test which is not included in commercial specifications is the cloud point. This feature should

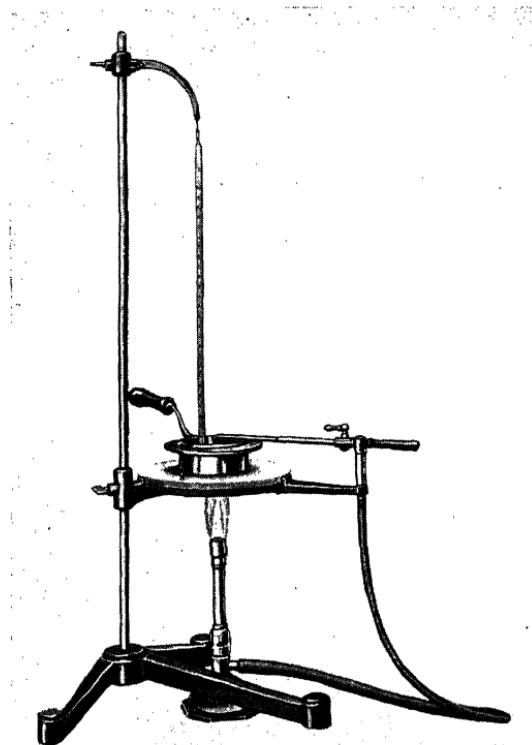


FIG. 10.—Cleveland open-cup flash and fire-test apparatus. (*Courtesy of Precision Scientific Company.*)

probably receive more attention than it has. The cloud point is defined by the American Society for Testing Materials as the temperature at which a paraffin wax or some other solid substance begins to crystallize out or separate from solution when the oil is chilled under definite prescribed conditions. It is prescribed that the test for cloud point shall be made only on oils which are transparent in layers $1\frac{1}{2}$ in. thick. For all other oils the pour point shall be used.

To again refer to the American Society for Testing Materials, the pour point is defined as the lowest temperature at which an oil will pour or flow when it is chilled without disturbance under definite prescribed conditions. This test is also called the cold test. The chief significance, and it is an important one, of this test is that it serves as an index of the suitability of an oil under certain climatic conditions, especially in the case of domestic and light commercial oils where preheating is not provided for liquid fuel. An oil cannot be pumped at temperatures lower than the pour point unless provision for heating the fuel is made.

CARBON, WATER, AND SEDIMENT TESTS

In an attempt to determine, if possible, the probable physical action of a fuel oil passing through a burner, as regards its tendency toward carbon formation, a test has been devised which is termed the Conradson carbon residue test. A sample of oil is subjected to destructive distillation in the absence of air, a strong heat being applied to it both while the vapors are being driven off and to the remainder after volatilization has ceased. The residue from this treatment is called the Conradson carbon residue.

It is customary to make a single determination of the matter present in an oil which can be separated by rapid whirling of the material in a centrifuge, and to report such matter as water and sediment (frequently referred to as B. S. & W.—bottom settling and water). This determination will measure all extraneous material, *i.e.*, all substances not dissolved in the petroleum product or homogeneous with it, which besides water will include mineral matter and sometimes asphaltenes. A separate deter-



FIG. 11.—Conradson carbon residue apparatus. (*Courtesy of Precision Scientific Company.*)

mination of mineral matter, or ash, by burning off all organic matter and driving out the water, will indicate how much of the above quantity is nonorganic, other than the water.

Color is not usually a specified quality, except in trade parlance which refers to colorless or water-white distillate, and then it is only a marketing consideration. Actually, color has little or no bearing on the burning characteristics of an oil. Color is not



FIG. 12.—Centrifuge for determining water and sediment (B.S. & W.). (*Courtesy of Precision Scientific Company.*)

mentioned at all in the American Society for Testing Materials or Bureau of Standards Commercial Standards.

HEATING VALUE OF FUEL OIL

On first thought it might seem that the determination of calorific value would be the most important test to be made on a fuel oil, since its ultimate value as a fuel must be in direct proportion to the heat available upon its combustion. But the facts are that fuel oils have a remarkably constant heat of combustion; the weight per unit volume, or specific gravity, determines this within very close limits. Several formulas have been promulgated to show the relationship between calorific value and specific gravity, the most recent being¹

¹ By Drs. Egloff and Morrell.

TABLE VII.—DETAILED REQUIREMENTS FOR FUEL OILS (Bureau of Standards, CS 12-40)

Grade*	Flash point, °F.		Water and sediment, per cent		Carbon residue, per cent		Ash, per cent		Distillation temp., °F.		Viscosity seconds			
	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	10 per cent point	90 per cent point	End point	Saybolt universal at 100°F.	Saybolt furol at 122°F.	
No. 1. Description of fuel oil														
1 Distillate oil for use in burners requiring a volatile fuel	100 or legal	165	0	Trace	0.05 on 10 per cent, residue	410	560 ^d			
2 Distillate oil for use in burners requiring a moderately volatile fuel	110 or legal	190	10 ^b	0.05	0.25 on 10 per cent residue	440	600					
3 Distillate oil for use in burners requiring a low viscosity fuel	110 or legal	230	20 ^b	0.10	0.15 straight	675	600 ^f	45			
5 Oil for use in burners requiring a medium viscosity fuel	130 or legal	1.00	800	50	40	
6 Oil for use in burners equipped with preheaters permitting a high viscosity fuel	150	2.03 ^g	6000	900	300	

Note.—Recognizing the necessity for low sulphur fuel oils used in connection with heat-treatment, nonferrous metal, glass and ceramic furnaces, and other special uses, a sulphur requirement may be specified in accordance with following table:

Grade of Fuel Oil

No. 1.....
No. 2.....
No. 3.....
No. 5.....
No. 6.....

Other sulphur limits may be specified by mutual agreement between the buyer and seller.

* It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

† Lower or higher pour points may be specified whenever required by conditions of storage or use.

For use in other than steam type baffle-flame burners carbon residue on 10 per cent residuum may be increased to a maximum of 0.12 per cent.

This limit may be specified by mutual agreement between the buyer and seller.

* If the maximum end point may be increased to 590°F. when used in burners other than sleeve type blue-flame burners.

† To meet certain burner requirements the carbon residue limit may be reduced to 0.15 per cent on 10 per cent residuum.

* The minimum distillation temperature of 600°F. for 90 per cent may be waived if A.P.I. gravity is 26 or lower.

* Water by distillation, plus sediment, by extraction. Sum, maximum 2.0 per cent. The maximum sediment by extraction shall not exceed 0.50 per cent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 per cent.

OIL BURNERS

TABLE VIII.—TOTAL (GROSS) AND NET HEATING VALUES OF PETROLEUM PRODUCTS*

(Accurate within 1 per cent)

Gravity		Density lb. per gal.	Total heat of combustion at constant volume, \dot{Q}_v			Net heat of combustion at constant pressure, \dot{Q}_p		
Degrees A.P.I., at 60°F.	Specific, at 60°/60°F.		Cal./g.	B.t.u./ lb.	B.t.u./ gal.	Cal./g.	B.t.u./ lb.	B.t.u./ gal.
10	1.0000	8.337	10,300	18,540	154,600	9,740	17,540	146,200
11	0.9930	8.279	10,330	18,590	153,900	9,770	17,580	145,600
12	0.9861	8.221	10,360	18,640	153,300	9,790	17,620	144,900
13	0.9792	8.164	10,390	18,690	152,600	9,810	17,670	144,200
14	0.9725	8.108	10,410	18,740	152,000	9,840	17,710	143,600
15	0.9659	8.053	10,440	18,790	151,300	9,860	17,750	142,900
16	0.9593	7.998	10,470	18,840	150,700	9,880	17,790	142,300
17	0.9529	7.944	10,490	18,890	150,000	9,900	17,820	141,600
18	0.9465	7.891	10,520	18,930	149,400	9,920	17,860	140,900
19	0.9402	7.839	10,540	18,980	148,800	9,940	17,900	140,300
20	0.9340	7.787	10,570	19,020	148,100	9,960	17,930	139,600
21	0.9279	7.736	10,590	19,060	147,500	9,980	17,960	139,000
22	0.9218	7.686	10,620	19,110	146,800	10,000	18,000	138,300
23	0.9159	7.636	10,640	19,150	146,200	10,020	18,030	137,700
24	0.9100	7.587	10,660	19,190	145,600	10,040	18,070	137,100
25	0.9042	7.538	10,680	19,230	145,000	10,050	18,100	136,400
26	0.8984	7.490	10,710	19,270	144,300	10,070	18,130	135,800
27	0.8927	7.443	10,730	19,310	143,700	10,090	18,160	135,200
28	0.8871	7.396	10,750	19,350	143,100	10,110	18,190	134,600
29	0.8816	7.350	10,770	19,380	142,500	10,120	18,220	133,900
30	0.8762	7.305	10,790	19,420	141,800	10,140	18,250	133,300
31	0.8708	7.260	10,810	19,450	141,200	10,150	18,280	132,700
32	0.8654	7.215	10,830	19,490	140,600	10,170	18,310	132,100
33	0.8602	7.171	10,850	19,520	140,000	10,180	18,330	131,500
34	0.8550	7.128	10,860	19,560	139,400	10,200	18,360	130,900
35	0.8498	7.085	10,880	19,590	138,800	10,210	18,390	130,300
36	0.8448	7.043	10,900	19,620	138,200	10,230	18,410	129,700
37	0.8398	7.001	10,920	19,650	137,600	10,240	18,430	129,100
38	0.8348	6.960	10,940	19,680	137,000	10,260	18,460	128,500
39	0.8299	6.920	10,950	19,720	136,400	10,270	18,480	127,900
40	0.8251	6.879	10,970	19,750	135,800	10,280	18,510	127,300
41	0.8203	6.839	10,990	19,780	135,200	10,300	18,530	126,700
42	0.8155	6.799	11,000	19,810	134,700	10,310	18,560	126,200
43	0.8109	6.760	11,020	19,830	134,100	10,320	18,580	125,600
44	0.8063	6.722	11,030	19,860	133,500	10,330	18,600	125,000
45	0.8017	6.684	11,050	19,890	132,900	10,340	18,620	124,400
46	0.7972	6.646	11,070	19,920	132,400	10,360	18,640	123,900
47	0.7927	6.609	11,080	19,940	131,900	10,370	18,660	123,300
48	0.7883	6.572	11,100	19,970	131,200	10,380	18,680	122,800
49	0.7839	6.536	11,110	20,000	130,700	10,390	18,700	122,200

* From *Miscellaneous Publications* 97 of U. S. Bureau of Standards.

For an uncracked product

$$\text{B.t.u. per lb.} = 17,600 + 69 \times \text{A.P.I. degrees}$$

For a cracked product

$$\text{B.t.u. per lb.} = 17,780 + 54 \times \text{A.P.I. degrees}$$

These formulas, as well as all tabular data and charts in this book which give heats of combustion, show the high or gross heating values, meaning that the water formed in combustion has been condensed and the heat of condensation of the water included in the heating value. In most processes, however, it is not practicable to recover this latent heat of condensation, and so, when the gross heating value is used in efficiency computations, the net efficiency is thereby reduced. This actual heat not recoverable from the combustion of hydrogen to water averages about 6 per cent of the gross heating value.

FUEL OIL SPECIFICATIONS

Table VII gives the latest specifications on the six grades of fuel oil now recognized by the American Petroleum Institute, U. S. Bureau of Standards and American Society for Testing Materials, as well as most firms and individuals concerned with the production, distribution and utilization of fuel oils. Table VIII contains data on calorific values of fuel oils.

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CHAPTER III

COMBUSTION OF FUEL OIL

Combustion is a chemical process. The complete chemistry of combustion is a fairly complex one, but fortunately it is not necessary to understand or consider it in full detail in order to grasp that much of the science which is essential to the proper utilization of oil fuel. This chapter will first present a simple review of the practical elements of the combustion process, then will apply these concepts to fuel oil itself, and will conclude with a more technical discussion of theoretical principles. The reader who does not care for formal theory may omit this last portion of the chapter. On the other hand, those who seek more elaborate treatment of the science of combustion, beyond what this chapter affords, will find a bibliography on the subject at the end of the chapter.

Of the various substances which may enter into combustion, the one indispensable is oxygen, so that it is logical to base a definition of combustion upon oxygen. Combustion, then, is the process of union of substances with oxygen, accompanied in most cases with the evolution of light and in all cases with the evolution of heat.

Regardless of the type of fuel, combustion involves only two principal chemical elements—carbon and hydrogen. Of the 92 elements known to chemistry, these two alone constitute the usable substance of fuels. A third element, sulphur, if present, will burn along with carbon and hydrogen, but it is not necessary to consider sulphur as a source of heat as it occurs only in a very small percentage in fuels. Except in a few specific cases, such as the oxyacetylene torch, oxygen for the combustion process is secured by introducing air into the combustion chamber, and since nitrogen constitutes 79.1 per cent of the atmosphere, its presence has an important bearing on the process. So it is possible to restrict a discussion of combustion to a consideration of these four major elements—carbon, hydrogen, oxygen and nitrogen, with sulphur as a possible fifth but minor element.

The chemical reactions and chemical changes taking place inside a firebox are so rapid that it is difficult to observe the performance of the reacting substances. But it is a relatively simple matter to secure samples of and to investigate the substances produced by combustion, so that most operating control is based upon analysis of the products of combustion. To illustrate, it will serve our purpose very nicely to follow an hypothetical volume of air as it successively enters the firebox, participates in combustion, and then emerges as flue gas.

For all practical purposes, air may be considered here as consisting of 20.9 per cent oxygen, O_2 , and 79.1 per cent nitrogen, N_2 , by volume. Figure 13 shows an hypothetical mole¹ of air, except that the air, of course, is a uniform and homogeneous mixture of gases; for the present discussion it is shown as though the two component gases have been segregated.

Now if such a thing were possible and this mole of air were to undergo change in *complete and perfect* combustion and the fuel used were pure carbon, C, it would appear as

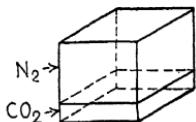


FIG. 13.—Mole of air.

flue gas with a composition as shown in Fig. 14. The nitrogen would be unchanged; the oxygen would have been quantitatively changed into carbon dioxide, CO_2 . As its temperature would undoubtedly be higher, the mole would occupy a greater space than when at room temperature.

But since perfect combustion cannot ordinarily occur, there would actually be some unused oxygen remaining in the flue gas, so that it would really appear as shown in Fig. 15 (still assuming that the fuel is *pure* carbon).

All commercial fuels, however, contain hydrogen in varying amounts. This hydrogen combines with some of the oxygen to form water, H_2O , which at the temperature of the combustion chamber, or of the chimney passages if above 212°F., is also a gas. Hence the flue gas from our mole of air is now as shown in Fig. 16.

It is important to note at this point that each molecule of oxygen, O_2 , forms two molecules of water, H_2O , so that the combustion of hydrogen from the fuel causes the total number of

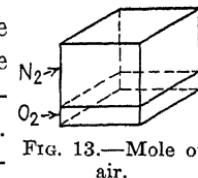


FIG. 14.—Mole of flue gas should pure carbon burn with complete and perfect combustion.

¹ A mole is 359 cu. ft. at standard conditions (see page 62).

molecules in the flue gas to become greater than the original number of molecules of the air which has been used. This same phenomenon occurs when carbon monoxide is formed. Hence the volume occupied by the flue gas illustrated in Figs. 16 and 17 is greater than the volume occupied by the mole of air which was the origin (aside, of course, from expansion which results from temperature rise, and which affects all gases equally). As a

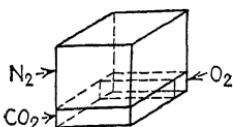


FIG. 15.—Mole of flue gas should pure carbon burn completely but with excess air.

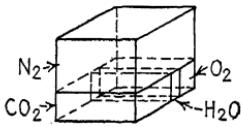


FIG. 16.—Mole of flue gas should hydrocarbon burn completely but with excess air.

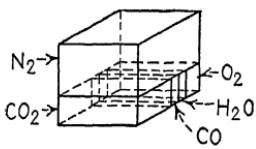


FIG. 17.—Mole of flue gas should hydrocarbon burn incompletely and with excess air.

result the 0.791 mole of nitrogen which entered with the air and which remains unchanged in amount although expanded from heat is now actually a smaller percentage of the total.

Hydrogen always burns to form water, there being no other possibility from the combustion of this element. In the case of

carbon, however, there are two possibilities. Besides carbon dioxide, carbon may form an intermediate product, carbon monoxide, CO, which is also a gas and which contains only one atom of oxygen per molecule, as mentioned above. When this is present in the flue gas (and the combustion therefore incomplete), the transformed mole must be pictured as in Fig. 17.

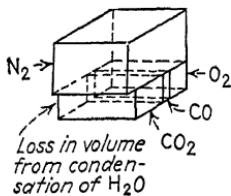


FIG. 18.—Mole of flue gas from incomplete combustion of hydrocarbon with excess air, after water of combustion has been condensed.

Of course, water, or rather steam, can thus be pictured in the flue gas only as long as its temperature remains above 212°F.

While this temperature is usually exceeded in the average breeching or chimney, the flue gas is brought down to room temperature when sampled and introduced into an Orsat apparatus, or other analyzing instrument. Immediately this occurs, the water condenses, and a decrease in volume of the products of combustion occurs, aside of course from that due to the contraction of the remaining (permanent) gases due to reduction in temperature. At room temperature, therefore, the gases appear as in Fig. 18.

In Fig. 19 is shown a more logical representation of these residual gases. The important difference now is that because of the condensation of water vapor each constituent gas represents a greater percentage than when water vapor was a component of the gas mixture. For instance, suppose the percentage composition, before condensation, to have been: nitrogen, 75.3; carbon dioxide, 11.3 per cent; oxygen, 3.7; carbon monoxide, 0.3; and water, 9.4. As soon as the water condenses, the gases which previously totaled only 90.6 per cent of the mixture now become 100 per cent. Accordingly, the new percentage composition is obtained by dividing each remaining constituent of the previous mixture by 0.906 to get the corresponding percentage of each constituent in the new mixture. Thus nitrogen is now present to the extent of $75.3/0.906$ or 83.1 per cent. Similarly, the percentages of the other gases are: carbon dioxide, 12.5; oxygen, 4.1; and carbon monoxide, 0.3.

It will be seen shortly that the apparent percentage of nitrogen in flue gas as indicated by an Orsat apparatus furnishes a valuable means of checking back other data, from the knowledge that the apparatus percentage results from the 79.1 per cent of the original air.

CALCULATIONS FROM FLUE-GAS ANALYSIS

It is now possible to draw some quantitative deductions from the flue-gas composition. It is apparent that when ordinary air is used for supporting combustion, when the fuel itself is pure carbon and when the combustion is to be complete and perfect, then the maximum possible carbon dioxide percentage in flue gas is obtained, and that percentage is 20.9. Under actual conditions, however, the attained percentage is lower because combustion is never perfect in that excess oxygen remains, combustion may not always be complete since carbon monoxide may be formed and the burning of hydrogen to form water introduces a product of combustion other than an oxide of carbon. Just what maximum attainable carbon dioxide is possible will be brought out as this chapter proceeds.

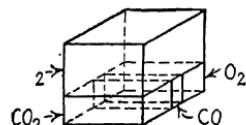


FIG. 19.—Mole of dry flue gas from incomplete combustion of hydrocarbon with excess air.

It is customary when using the complete Orsat apparatus to determine the percentage of carbon dioxide and oxygen in flue gas and then to take the difference between their sum and 100 per cent as the percentage of nitrogen. Some types of gas analysis apparatus provide an absorption tube for carbon monoxide also, but the determination of carbon monoxide is a matter of some analytical difficulty and it is doubtful if an Orsat can give a dependable reading on this. It will be shown later that the occurrence of carbon monoxide in flue gases from modern oil burners is very unlikely, so it will be assumed that ordinarily it is not present. For checking most oil-combustion processes it is sufficient then to determine carbon dioxide and oxygen and to figure nitrogen by difference.

Suppose such a case where the analytical results are: carbon dioxide, 12.5 per cent; oxygen, 4.1 per cent; and nitrogen, 83.4 per cent. From these results numerous important calculations can be made.

It is known that the percentage composition of air is 20.9 oxygen and 79.1 nitrogen. Since it is manifest that 100 moles of flue gas must contain 83.4 moles of nitrogen and that this nitrogen must have come from air which contains only 79.1 moles of nitrogen per 100 moles of air, it follows that it would require 83.4/0.791, or 105.4, moles of air to give 100 moles of flue gas.

There is a total of 16.6 moles of oxygen in the 100 moles of dry flue gas—12.5 as carbon dioxide and 4.1 as oxygen. But the 105.4 moles of air which formed this 100 moles of flue gas contained 105.4×0.209 , or 22.0, moles of oxygen. The difference, which is $22.0 - 16.6$, or 5.4, must have combined with hydrogen to form water, which of course has condensed out and does not appear in the analysis of the dry flue gas. This 5.4 moles of oxygen as water has combined with 10.8 moles of hydrogen.

It is now possible to tell something of the fuel oil which has been burned. The only carbon in the flue gas is the 12.5 moles occurring as the dioxide. By multiplying 12.5 moles by the atomic weight of carbon, which is 12, we find that 150 lb. of carbon were burned per 100 moles of dry flue gas produced, and similarly, by multiplying 10.8 moles of hydrogen by the molecular weight (2.02), it follows that 21.8 lb. of hydrogen were consumed. The total weight of the oil burned per 100 moles of flue gas was therefore $150 + 21.8$, or 171.8; by composition it contained

150/171.8, or 87.3, per cent carbon and 12.7 per cent hydrogen.¹

One other deduction concerning the fuel oil can be made. There are 12.5 moles or atoms of carbon and 2×10.8 , or 21.6, atoms of hydrogen. The ratio 12.5/21.6 is 0.58, which shows that the carbon-to-hydrogen relationship is greater than would be indicated by C_nH_{2n} ; i.e., that there must be present some hydrocarbons other than the olefin series.

Returning to the flue gases again, one of the most important deductions to be made from the analysis is the percentage of excess air. This is most conveniently made from the oxygen balance since, of course, the oxygen comes from air and therefore a quantitative statement concerning either is applicable to the other. The necessary oxygen is that which has entered into combustion—the unnecessary being that which remains uncombined in the flue gases; the ratio of the two gives the percentage of excess.

The unnecessary oxygen is shown by the flue-gas analysis to be 4.1 per cent. The necessary oxygen is that which has combined with carbon and hydrogen, and for this case is 12.5 + 5.4, or 17.9, per cent. The percentage of excess oxygen, and hence of excess air, is therefore established as 4.1/17.9, or 22.9 per cent.

So far nothing has been said of the amount of heat liberated in a combustion reaction, nor has there been discussed the relationship of gas volume changes with respect to temperature and pressure. To understand these subjects, it is necessary to consider, in brief, the so-called gas laws.

The gas laws are a group of physical and chemical statements which set forth the characteristics of gases and their behavior under varying physical and chemical environments. Inherently, these laws are mathematical in that they express quantitative relationships, but in this discussion the mathematics will be restricted to an elementary nature.

BOYLE'S LAW

This is probably the most familiar of the gas laws. It states that, at constant temperature, the volume of a gas is inversely pro-

¹ Strictly, what is here called weight of oil, carbon or hydrogen, should be termed combustible, per cent carbon and per cent hydrogen in the combustible, respectively.

portional to the pressure. A cubic foot of air, for instance, at atmospheric pressure (14.7 lb. per sq. in. or 76 cm. of mercury at sea level) will occupy a volume of $\frac{1}{2}$ cu. ft. at a pressure of 2 atm. or 29.4 lb. per sq. in., or of $\frac{1}{3}$ cu. ft. at a pressure of 3 atm., 44.1 lb. per sq. in. Conversely, the cubic foot of air or any other gas, at atmospheric pressure, will expand to a volume of 2 cu. ft. if the pressure diminishes to 7.35 lb. per sq. in.

It must be clearly understood that the pressures referred to in the preceding paragraph are absolute pressures, and have no reference to pressures indicated on the ordinary pressure gauge. Such instruments ordinarily indicate pressures above or below atmospheric, *i.e.*, the zero point on a pressure gauge is 14.7 lb. abs. Hence 2 atm. abs. would be indicated on a pressure gauge as 14.7 lb. pressure, while $\frac{1}{3}$ atm. abs. would show on a pressure gauge as 20 in. of vacuum.

CHARLES'S LAW (ALSO LAW OF GAY-LUSSAC)

At constant pressure, all gases expand or contract with temperature increase or decrease, respectively, at a rate of $\frac{1}{460}$ of the volume at 0°F. for each degree Fahrenheit by which the temperature rises or falls.

To illustrate, assume 1 cu. ft. of gas at 0°F. to be heated. At a temperature of 23°F. the volume has increased by $\frac{23}{460}$, or $\frac{1}{20}$. At 230°F. the volume has increased by one-half, and at 460°F. the volume has doubled. Similarly, if the gas be cooled, at -50°F. the gas has lost $\frac{50}{460}$ of its initial volume; at -460°F. the volume would have been reduced to zero.

It is common practice, when correcting gas volumes for pressure and volume, to make a simultaneous computation by combining both laws—that of Boyle and that of Charles. The rule is: Multiply the initial volume by the ratio of the final absolute temperature to the initial temperature, and then multiply this result by the ratio of the initial pressure to the final pressure.

Before going into examples of applications of this rule, the following points should be brought out. Absolute temperature is secured by adding the Fahrenheit temperature to 460 (the absolute zero on the scale being -460). Pressures may be expressed in any units, since a ratio is independent of units. So long as both initial and final pressures are expressed in the same

units, it matters not if pressures be expressed as atmospheres, pounds, inches of water, centimeters of mercury, etc.

The following equation summarizes this rule:

$$v = v_0 \times \frac{T}{T_0} \times \frac{p_0}{p}$$

in which v = final volume.

v_0 = initial volume.

T = final temperature, °F. + 460.

T_0 = initial temperature, °F. + 460.

p = final pressure.

p_0 = initial pressure.

Example.—Under atmospheric conditions of 78°F. and 29.9 in. of mercury, air is entering a combustion chamber at the rate of 100 c.f.m. If the stack temperature is 560°F. and the pressure in the stack 0.5 in. below atmospheric, how many cubic feet of nitrogen are passing through the stack per minute?

Solution.—100 cu. ft. of air contain 79.1 cu. ft. of nitrogen. Hence the initial conditions are: volume, 79.1; temperature, 538; pressure, 29.9. The final temperature is 1020 and the final pressure is 29.4. Hence

$$\begin{aligned} v &= 79.1 \times \frac{1020}{538} \times \frac{29.9}{29.4} \\ &= 152.5 \text{ cu. ft.} \end{aligned}$$

Example.—Pure oxygen is to be stored in a steel cylinder. The initial temperature is 68 degrees at atmospheric pressure. If 985 cu. ft. are compressed into a cylinder having a cubical content of 94 cu. ft., and if the tank and contents are cooled down to 40°F., what is the pressure in the steel tank?

Solution:

$$\begin{aligned} 94 &= 985 \times \frac{500}{528} \times \frac{1}{p} \\ p &= \frac{985 \times 500}{94 \times 528} = 9.92 \text{ atm.} \\ &= 145.8 \text{ lb. per sq. in. (at sea level)} \end{aligned}$$

In working problems of volume change, should there be any confusion as to the proper arrangement of the factors in the formula, it will help to bear in mind the logical consequences of pressure and temperature changes. An increase in temperature will expand a gas, so that the ratio of the temperatures must then be greater than unity; should the temperature fall the ratio must

be less than one. Similarly a gas will be compressed by increase of pressure, so that the ratio of pressures is a proper fraction. The converse, of course, requires an improper fraction to show an expansion of a gas with reduction of pressure.

AVOGADRO'S HYPOTHESIS

In the earlier discussions of this chapter it was pointed out that each percentage of oxygen in air could reappear as a percentage of carbon dioxide in flue gas, but that a percentage of oxygen forming carbon monoxide would create 2 per cent of carbon monoxide. These statements anticipated, and are correct because of the truth of Avogadro's hypothesis, which states: *At the same conditions of pressure and temperature, equal volumes of all gases contain equal numbers of molecules.*

The exact number of molecules in a given volume of gas cannot easily be determined, nor is it necessary or important to know the number in order to apply this law. A most useful derivation from the law is the conception of the molal gas volume.

A molal gas volume is a quantity of a gas which weighs the number of mass units indicated by its molecular weight. For the common gases, the molecular weights, or molal weights, are:

Gas	Symbol	Molal weight	Gas	Symbol	Molal weight
Oxygen.....	O ₂	32	Ethane.....	C ₂ H ₆	30
Nitrogen.....	N ₂	28	Carbon dioxide....	CO ₂	44
Hydrogen.....	H ₂	2	Carbon monoxide..	CO	28
Methane.....	CH ₄	16	Water.....	H ₂ O	18

From Avogadro's hypothesis it follows that the volume occupied by the formula weight or molal weight of all gases must be the same. This volume is expressed in scientific work (when the formula weight is expressed in grams) as 22.4 liters at a temperature of 0°C. and a pressure of 760 mm. of mercury. In engineering units (with the formula weight in pounds), this molal volume or mole as it will be referred to in this book, is 359 cu. ft. at 32°F. and 29.9 in. of mercury.

It is now possible to return to our combustion problem and deduce some additional conclusions concerning the quantitative relations of the reacting substances.

It has been shown above that per 100 moles of flue gas there were required 105.4 moles of air and the fuel burned was 171.8 lb. This can be converted into an expression for cubic feet of air per pound of oil by applying the gas laws. Dividing, we secure moles of air per pound of oil, $105.4/171.8$, or 0.613. Since a mole at standard conditions occupies 359 cu. ft., 0.613 mole will be 220.1 cu. ft. at 32°F. and 29.9 in. of mercury.

Now suppose the temperature of the air entering the firebox to be 78°F. and barometric pressure to be 30.3 in. of mercury. Then the volume of air used per pound of fuel oil is

$$220.1 \times \frac{460 + 78}{460 + 32} \times \frac{29.9}{30.3} = 237.4 \text{ cu. ft.}$$

It is also possible to calculate the volume of total flue gases produced per pound of fuel burned. For each 100 moles of dry flue gas there are also 10.8 moles of water vapor, which makes a total 110.8 moles of total flue gas to 171.8 lb. of fuel, or 0.645 mole per lb. If the stack gas temperature is 490°F. the flue-gas volume per pound of fuel is

$$0.645 \times 359 \times \frac{460 + 490}{460 + 32} = 447.1 \text{ cu. ft. per lb. oil}$$

There remains one more result to be figured from the flue-gas analysis. If the gross calorific value of the fuel is 18,300 B.t.u. per lb., and if the heat of condensation of water is 970.4 B.t.u. per lb., then the percentage of the gross fuel value lost through noncondensation of the water vapor formed in combustion is evaluated as follows:

Ten and eight-tenths moles of water vapor formed per 171.8 lb. of oil burned gives 0.063 mole per lb., and since the molal weight of water vapor is 18, the weight of water vapor formed per pound of fuel burned is 1.13 lb. The percentage of heat loss is then

$$\frac{1.13 \times 970.4}{18,300} \times 100 = 6.0 \text{ per cent}$$

THERMAL CALCULATIONS FROM FLUE-GAS ANALYSIS

This chapter so far has considered the chemical and physical relationships between air, flue gas and fuel oil and, except for the last computation above, has ignored heat relationships entirely.

In order to develop the technology of heat release and combustion efficiency of fuel oil it will first be necessary to discuss some additional physical and chemical concepts and theories.

The science of heat itself is extensive and complex; for the purposes of the present work it is sufficient to say that heat is a form of energy and that in general it is manifested by temperature phenomena. Thus the addition of heat to a substance will raise its temperature—unless a change of state occurs. The removal of heat from a substance will lower its temperature unless a change of state occurs. By change of state is meant the transition from solid to liquid or liquid to gas, or vice versa, or change from one crystalline form to another or to an amorphous condition, or the reverse. It is not the intention of this book to consider the *nature or characteristics* or complete *utilization* of heat; rather it will concern itself with only the *efficient liberation of heat from fuel oil*.

There are two definite concepts to keep in mind relative to the measurement of heat. These might be termed qualitative and quantitative concepts. The first is described as temperature; the second as heat units. Both are familiar terms, so that there should be no cause for confusion between them. In this book temperatures will generally be expressed on the Fahrenheit scale, and heat units will generally be given as British thermal units (B.t.u.). In scientific work, the centigrade scale and the calorie are the universally used analogues.

Temperature indicates the degree or *intensity* of heat, much the same as volts give electrical intensity or feet of head give hydraulic intensity. British thermal units give *amount* of heat, in the same way that amperes give amount of electrical energy and gallons give amount of potential hydraulic energy at any stated head. Another analogue would be had in the case of steam; a given pressure tells the intensity of any quantity of weight expressed in weight units; the term pounds may be used in connection with both, but for pressure, pounds expresses a force intensity—for weight, or amount, pounds implies a gravitational attraction.

To establish the Fahrenheit scale, the freezing point of *pure* water is taken as 32°, and the boiling point of pure water is 212°. The interval between them is divided into 180 equal parts and each is termed 1°F. Extension is then made below the 32 mark

and above the 212 mark using the unit which has thus been established. Thus the zero point is 32° under the freezing point; absolute zero is 460°F. below zero.

A B.t.u. is $\frac{1}{180}$ part of the heat required to raise 1 lb. of water in temperature from 32°F. to 212°F. Commonly, a B.t.u. is said to be the heat to raise 1 lb. of water through 1°F. The two are very closely equal. A B.t.u. is equal to 252 calories.

In Chap. II it was shown that the ratio of the weights of unit volumes of water and any other substance is called the specific gravity of the second substance. In a similar way, the ratio of the heat required to raise the temperature of a unit weight of any substance to the heat required to raise the temperature of the same weight of water through the same range is called the specific heat of the substance. It follows, of course, that the specific heat of water itself is unity, or one. In the Appendix is a table of specific heats of common solids and liquids.

In defining the specific heat of a gas it is necessary to postulate another condition, *viz.*, the mode of behavior of the gas as the temperature is raised. From the gas laws it is known that the addition of heat to a gas will tend to raise its pressure, or volume, or both. Because of thermodynamic principles, it is necessary, when giving the specific heat of a gas, to state whether the gas pressure remains constant by allowing the gas to expand as the temperature rises, or whether the volume is maintained constant by permitting the pressure to increase with the temperature. The first is known as the specific heat at constant pressure and is designated as C_p . The second is called the specific heat at constant volume, and is designated by the symbol C_v . In general, when specific heats of gases are referred to or used in this book, it will be understood to be C_p , unless otherwise stated. The chart of Fig. 20 shows the mean specific heats of the gases encountered in combustion work. This chart gives the molal specific heat, or B.t.u.'s required to raise the temperature of 1 mole of the gas through 1°F. Since the value of C_p changes with the temperature, to facilitate calculations, this chart is based upon the average of all the values of C_p between 60°F. and the indicated temperature on the abscissa scale. Should the initial gas temperature be other than 60°F. , very little error will be introduced by using these values, provided it lies within the ordinary range of room temperatures.

The specific heat of a mixture of gases is found by multiplying the C_p of each constituent of the mixture by its respective percentage, and adding the totals. Thus for the flue gas which has

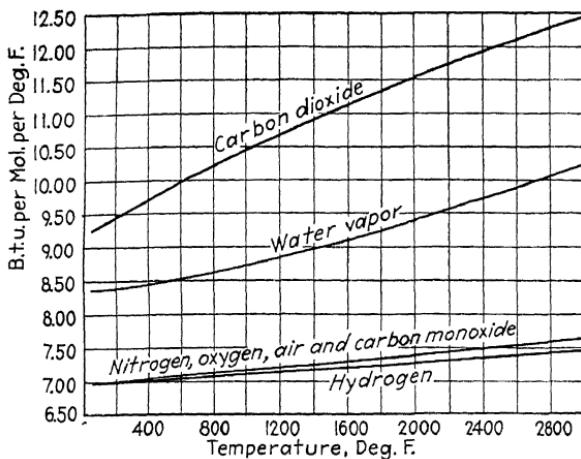


FIG. 20.- Specific heats of common gases (molal).

been considered on page 57, and which has the composition: nitrogen, 75.3 per cent; carbon dioxide, 11.3 per cent; oxygen, 3.7 per cent; carbon monoxide, 0.3 per cent; and water, 9.4 per cent; the specific heat to be used in calculations would be determined by the following computation:

Gas	Molecular weight (MW)	Molal C_p	Percentage (P)	$MW \times P$	$C_p \times P$
Nitrogen.....	28	7.06	0.753	21.08	5.32
Carbon dioxide.....	44	9.84	0.113	4.97	1.11
Oxygen.....	32	7.06	0.037	1.18	0.26
Carbon monoxide.....	28	7.06	0.030	0.84	0.21
Water.....	18	8.53	0.094	1.69	0.80
Averages.....	29.76	7.70

It is thus seen that the molecular weight of the flue gas is 29.76, and that its average or mean molal specific heat is 7.70. If it be desired to know the specific heat in B.t.u. per degree per pound of flue gas, this is very simply obtained by dividing the molecular weight into the molal C_p , to get 0.259 B.t.u. per °F. per

lb. Or again, since a mole contains 359 cu. ft. at standard conditions, the mean specific heat in B.t.u. per degree Fahrenheit per cubic foot at 32°F. and 29.9 in. would be 7.70/359, or 0.0215. (It should be remembered that all these expressions for specific heat are mean values over the temperature range involved: room temperature to 490°F.)

It is now possible to discuss heat release in quantitative terms, and to determine upon both theoretical and practical results desired in the combustion process.

The calorific value of fuel is determined by burning a measured quantity under conditions which will permit of observing the temperature rise of the surrounding system; from a knowledge of the heat capacity of that system a calculation is made of the amount of heat which has been released by the burning of the fuel.

It is interesting to note that the theoretical amount of heat which can be liberated from a substance by its combustion is in no way affected by the speed of the combustion process. Thus if a piece of coal be permitted to weather by exposure to air and dampness at ordinary temperatures, a slow combustion takes place, whereby, over a considerable period of time, the carbon changes to an oxide and the hydrogen to water. The amount of heat so released is identical with that given out when the same weight of the same coal is burned instantly in an incandescent firebox to the same products of combustion.

In spite of the foregoing statement, however, there can be, and in actual practice there exists, a vast difference between the theoretical available heat and that actually secured in useful heat from the combustion process. The fault is not with the theory, for the available heat is actually released. The fault lies rather in the system in which the fuel is burned, and it is toward the theoretical performance in such systems that we must now direct our attention.

The most conspicuous phenomenon accompanying combustion is flame or, in the case of some fuels like coke, incandescence. In general, a flame is the result of combustion of gases, and may be luminous or nonluminous. The luminosity is merely due to the presence in the flame of carbon particles which have been heated to the temperature of incandescence so that they glow. Similarly, a bed of burning coal from which the volatile matter has been driven (and which is therefore coke) will glow because of the light

emitted by incandescent carbon. Chemically, a luminous fire is generally termed oxidizing and the nonluminous fire reducing. The luminosity has no bearing upon the flame temperature attained or upon the amount of heat released if combustion in both cases is complete. But it should be noted that the luminous fire is more prone towards smoking and carbon deposition than is the reducing fire. On the other hand, the reducing fire is more characteristic of burning gas than of burning oil.

The theoretical flame temperature is the temperature that would be attained if all the heat resulting from combustion were to be utilized in raising the temperature of the products of combustion. From this it follows that the theoretical flame temperature depends upon the initial temperature of the fuel and air, for naturally if the initial temperature is 500°, the resulting temperature rise will carry the products of combustion to a higher temperature than if the fuel and air were at 100° when entering the combustion chamber.

If the temperature rise be referred to a fixed point, say 32°F., then the flame temperature attained above that point will be theoretically:

$$T = \frac{H_c + H_f + H_a}{W_g \times C_p}$$

in which T = °F. above 32, flame temperature.

H_c = sensible heat of combustion, i.e., less heat of condensation of water formed.

H_f = sensible heat of the fuel above 32°F.

H_a = sensible heat of the air above 32°F.

W_g = weight of products of combustion (flue gases).

C_p = mean specific heat of the flue gases.

Actually, of course, the theoretical flame temperature is never attained, some of the principal causes being radiation of heat from the flame, for a hot body is constantly radiating heat to its environment at a rate which is proportional to the fourth power of the temperature difference; loss by conduction from the flame into any body with which the flame is in contact; loss by incomplete combustion.

It should be noted that, as in any fraction, the greater the value of the denominator the lower the value of the quotient. Hence, if the weight of the products of combustion be abnormally large

through the presence of large quantities of excess air, the flame temperature is correspondingly reduced.

It will be seen in later chapters that a large amount of the heat of combustion is transferred into the boiler or furnace by radiation from the flame and refractory firebox, so that, in general, efficiency is enhanced by high flame temperatures. Excess air, therefore, in lowering the flame temperature lowers the rate of heat transferred by radiation. But excess air causes a much greater heat loss by carrying away sensible heat, for all the oxygen and nitrogen which pass through the firebox without chemical change (and of course none of the nitrogen undergoes such) have to be heated from entering temperature to exit temperature.

On page 63 it was shown that 0.645 mole of flue gas was formed per pound of fuel oil and that the temperature increased from 78 to 490°F. Then the sensible heat carried away in the flue gases must be $0.645 \times (490 - 78) \times 7.7 = 2045$ B.t.u. per lb. of fuel. Since the calorific value of the fuel is 18,300 B.t.u. per lb., the percentage of heat lost as sensible heat of the stack gases is $(2045/18,300) \times 100 = 11.2$.

THEORETICAL ASPECTS OF COMBUSTION

Thus far in this chapter, there have been presented matters which concern the practical interpretations of chemical and physical laws as they affect combustion—practical to the extent of showing how efficiently oil is being burned, how much heat is lost up the stack, what portion of that is avoidable, etc. The remaining theories of science pertaining to combustion are less practical, in that they attempt to describe, not so much the results of combustion, as the *mechanism* of combustion from the scientific viewpoint. These matters are of academic interest only, but for the sake of completeness, a brief review is here presented of the more prominent theories.

Oil, of course, burns only when in the gaseous phase; from the liquid state, it must be first evaporated into a vapor or be mechanically atomized into a fine spray approaching a vapor in its state of attenuation. Methods of atomizing fuel oil are considered in detail in Chaps. V and VI; it is merely necessary to note here that oil as a liquid will not burn. The second essential is that the fuel be at a temperature equal to or above its ignition temperature, for every fuel has a characteristic ignition temperature which is

the lowest temperature at which it can be ignited and supported in combustion.

Although the combustion process is an extremely rapid reaction under ordinary firing conditions, it will be recalled that combustion is only one type of oxidation reaction and that oxidations in

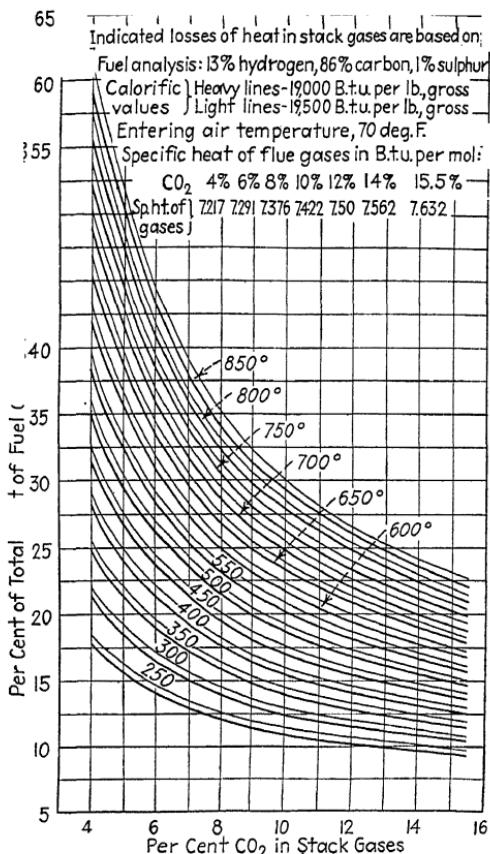


FIG. 21.-Loss of heat in stack gases for light fuel oils.

general can be controlled. When oxygen is introduced slowly into the hydrocarbon molecule a series of chemical compounds are formed (Chap. I). Thus it has been seen that the successive oxidation products starting from ethane, C_2H_6 , are: ethyl alcohol, $\text{C}_2\text{H}_5\text{OH}$; acetaldehyde, $\text{C}_2\text{H}_4\text{O}$; acetic acid, $\text{C}_2\text{H}_4\text{O}_2$; and finally carbon dioxide and water. Now it has been proved that all these intermediate products of partial oxidation are actually formed

in the firebox of an oil fire, and because the entrance of the first atom of oxygen into the hydrocarbon molecule forms the hydroxyl radical, OH, the theory is called the hydroxylation theory.

Actually, once oil has passed from the liquid to the vapor phase, it becomes a gaseous fuel and its theoretical behavior thereafter

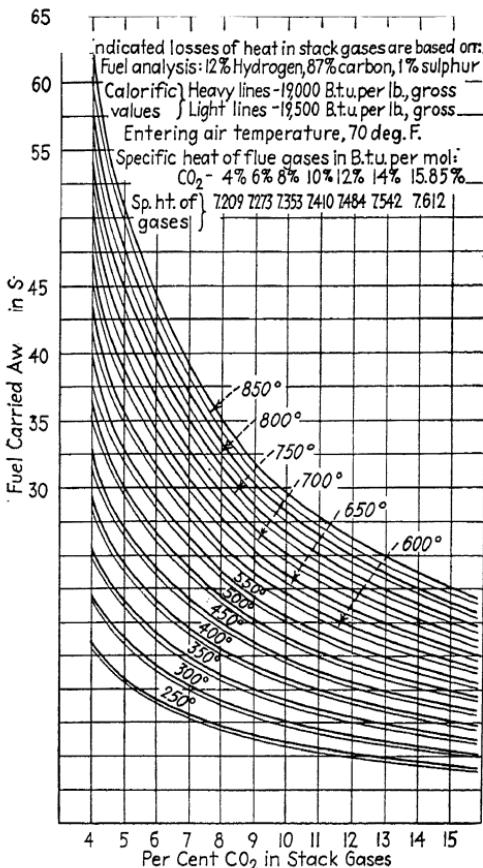


FIG. 22.—Loss of heat in stack gases for medium fuel oils.

is identical with the underlying theory for all fuel gases. Most fuel gases, however, contain free hydrogen and carbon monoxide, whereas neither of these is present in a vaporized hydrocarbon unless decomposition occurs. If the burner design is correct, and if the atomization of oil and the mixture of the spray with oxygen are adequate and successfully accomplished, then hydroxylation should occur and normal combustion result. But, should either

the atomization be incorrect or the subsequent admixture with air be improper, the hydrocarbons will crack or thermally decompose when subject to the heat of a firebox, with the resultant formation of soot from free carbon and of hydrogen and carbon monoxide which will burn with a smoky fire.

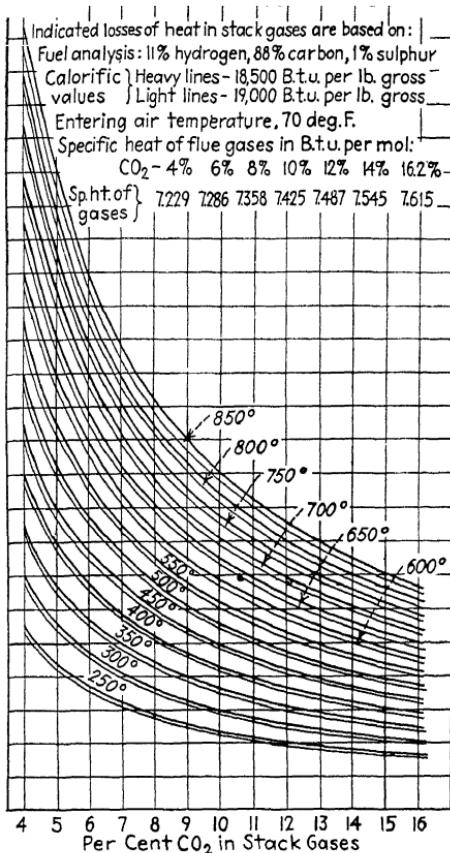


FIG. 23.—Loss of heat in stack gases for heavy fuel oils.

Prior to the hydroxylation hypothesis, experimenters believed that oxygen sometimes and under certain conditions exhibited a stronger affinity for hydrogen than for carbon, and that under other conditions the reverse was true. Therefore, when the environment favored the union of oxygen with hydrogen in preference to carbon, a slight deficiency of air would result in smoke and carbon formation. Under the hydroxylation theory, of course, it

is clear that oxygen is not called upon to show any favoritism at all—but that if an oil burner of good design is functioning with the proper air supply and with a firebox at the right temperature, then the hydrocarbon is oxidized uniformly, with the simultaneous formation of carbon dioxide and water.

The following formulas offer a means of calculating the theoretical quantity of air required to effect perfect combustion of various types of hydrocarbons. They are predicated on the relationship that 1 lb.-mole of oxygen is contained in 1820 cu. ft. of air at 60°F. and 29.9 in. In each case, the volume V is in cubic feet under the same conditions, and is the volume of air needed per pound of hydrocarbon.

Paraffins C_nH_{2n+2}	Olefins and Naphthenes C_nH_{2n}
$V = \frac{3 + \frac{1}{n}}{\frac{4}{28} + \frac{1}{n}}(1820)$	$V = \frac{3}{28}(1820) = 195$
Acetylenes and Diolefins C_nH_{2n-2}	Olefin-acety- lenes C_nH_{2n-4}
$V = \frac{3 - \frac{1}{n}}{\frac{4}{28} - \frac{4}{n}}$	$V = \frac{3 - \frac{2}{n}}{28 - \frac{8}{n}}$

It should be noted that as the molecular weights increase, the required air approaches that for the olefins.

It is perhaps fitting to bring this chapter to a close by pointing out a sharp distinction between normal oil combustion and normal coal combustion. Whether coal be burned upon grates or in a stoker, the almost universal method of supporting combustion is to direct a stream of air in some manner through a bed of incandescent coke, which bed has been created through the distillation of the volatile matter from the green coal. Now oxygen upon coming in contact with glowing coke immediately forms carbon dioxide. As the dioxide ascends or travels farther through the coke bed, however, the action of passing over additional glowing coke in an atmosphere which has been depleted of oxygen causes a reduction of the dioxide by the hot carbon to

form monoxide. The monoxide leaves the fuel bed as a gas, and must be supplied with additional oxygen from a secondary air supply in order that it shall burn. Failure to supply adequate secondary air, or to bring air into intimate mixture with the monoxide from the fuel bed results in heat loss through incomplete combustion. Hence the normal combustion of coal induces a preliminary formation of carbon monoxide, which can only be burned by a secondary combustion process in the space above the fuel bed.

With fuel oil, as has been explained, the normal combustion of hydrocarbons avoids entirely the formation of carbon monoxide. It is only improper combustion, which induces cracking and subsequent formation of carbon monoxide, that can cause incomplete combustion of fuel oil. Hence, when an oil burner is operating properly, it is very unlikely that carbon monoxide will be found in the stack gases.

SUMMARY OF EQUATIONS OF CALCULATIONS

In the following equations, the percentages of carbon dioxide, oxygen and nitrogen in the flue gas, as determined by Orsat, are used as whole numbers. For instance, if the Orsat gave a reading of 12.3 per cent carbon dioxide in a particular case, then in place of CO_2 in the following equations, the number 12.3 would be substituted.

1. Moles of air used per 100 moles of flue gas = $\frac{N_2}{0.791}$

2. Total weight of oil burned per 100 moles flue gas

$$= 7.96\text{CO}_2 - 4.04\text{O}_2 + 1.07\text{N}_2$$

3. Percentage of carbon in fuel oil

$$= \frac{\text{CO}_2}{7.96\text{CO}_2 - 4.04\text{O}_2 + 1.07\text{N}_2}$$

4. Excess air percentage used = $\frac{\text{O}_2}{0.264\text{N}_2 - \text{O}_2} \times 100$

5. Moles of wet flue gas per pound oil burned

$$= \frac{100 + 0.528\text{N}_2 - 2\text{CO}_2 - 2\text{O}_2}{7.96\text{N}_2 - 4.04\text{O}_2 + 1.07\text{N}_2}$$

6. Moles of air used per pound of oil burned

$$= \frac{\text{N}_2}{6.39\text{CO}_2 - 3.2\text{O}_2 +}$$

Cubic feet of air at standard conditions used per pound oil burned = moles of air \times 359

7. Quantity of heat lost by noncondensation of water in flue gases, B.t.u. per pound oil burned

$$= \frac{0.528N_2 - 2CO_2 - 2O_2}{7.96CO_2 - 4.04O_2 + 1.07N_2} \times 17,367$$

8. Quantity of heat lost as sensible heat of flue gases, B.t.u. per pound oil burned

$$\frac{100 + 0.528N_2 - 2CO_2 - 2O_2}{7.96CO_2 - 4.04O_2 + 1.07N_2} \times (T_s - T_r) \times C_p$$

in which T_s = temperature of flue gases.

T_r = temperature of entering air.

C_p = molal specific heat of flue gases.

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CHAPTER IV

COMPARATIVE FUEL DATA

DATA ON SOLID FUELS

Coal is defined by the 1934 Webster's International Dictionary as "a black, or brownish black, solid, combustible mineral substance formed by the partial decomposition of vegetable matter without free access of air, under the influence of moisture and, in many cases, of increased pressure and temperature. A complete series can be traced from the cellulose of wood through lignite, or brown coal, and soft, or bituminous coal, to hard coal or anthracite or, as a final product, to graphite. The order given is one of decreasing volatility and increasing carbon content. Besides carbon, coal contains hydrogen, oxygen, nitrogen, and sulphur, and also inorganic constituents, which last are left behind as ash when the coal burns. The largest deposits of coal belong to the Carboniferous system, but the coal of the Cretaceous system (Laramie series), is also very extensive."

There is no universal system or method of classifying coals, but practically all of the most important systems base the classification upon the "proximate analysis." The proximate analysis is determined from a procedure standardized by the American Society for Testing Materials. *Moisture* is determined by drying a weighed sample at 105°C. (221°F.) for 1 hr.; the loss in weight is called moisture. Similarly, *ash* is determined by the weight of what remains after the coal is burned under specified conditions, *volatile combustible* is the loss in weight on heating the coal 7 min. at 950°C. (1742°F.), less the moisture, and the coke residue from the volatile procedure, less the ash, is the *fixed carbon*.

The classifications of coals used by the U. S. Geological Survey and U. S. Bureau of Mines are shown on p. 77.

Attention must be directed to the fact that in the preceding table the indicated values are on an *ash-free* basis. It is necessary to use this basis when trying to thus establish a system of coal classification, in order to eliminate mineral matter from the

TABLE IX.—CLASSIFICATION SYSTEM USED BY U. S. GEOLOGICAL SURVEY AND U. S. BUREAU OF MINES

Rank	Equivalent term	Proximate analysis on ash-free basis, per cent			Calorific value on ash-free basis, B.t.u. per lb.	Physical properties and occurrence
		Moisture	Volatile matter	Fixed carbon		
A	Peat.....	90-80				
B	Lignite.....	50-32	25-30	28-37	6,400- 8,300	Slacks freely on weathering; noncoking; northern part of Great Plains and Gulf coastal area
C	Subbituminous...	32-12	32-38	36-50	8,550-11,800	Slacks considerably on weathering; noncoking; western states and possibly certain coal fields of Iowa, Ill., Mo., western Ky. and Ind.
D	Bituminous low rank.....	12-5	39-42	49-53	12,000-13,600	Slight or no slacking on weathering; may be coking; all coal-bearing states except the Dakotas
E	Bituminous high rank.....	5-35	40-22	55-75	14,000-15,000	No slacking properties; often coking; most abundant in Appalachian field but found in most coal-bearing states
F	Semibituminous,.....	26-23	23-15	74-82	15,200-15,300	Friable; coking; nearly smokeless; central Pa.; New River Pocahontas field; Fort Smith field of Ark. and Okla.
G	Semianthracite...Fuel ratio (F.C./V.M.*), from 4.0 to 10.0					Less hard and less lustrous than typical anthracite; near Lykens, Bernice and Carbondale in Pa.; Meadow Branch field of W. Va.; and Valley field of Va.
H	Anthracite.....	All Pennsylvania anthracite with fuel ratio over 10.0				Pa.
I	Superanthracite.....					Resembles graphite; R. I. and southern Mass.

* F.C. = fixed carbon; V.M. = volatile matter.

OIL BURNERS

analysis of the progressive changes which are exhibited by the organic matter which forms the pure coal substance. But in actual practice, the value of coals is determined by their "as received" analyses, which show just what the coal purchaser receives and what the coal-burning plant must use as fuel.

Besides the proximate analysis which has been above described, there is also an ultimate analysis of coal which shows exactly the percentage composition of the coal in terms of the chemical elements present in the dry coal. Following are typical data on representative American coals.

TABLE X.—AVERAGE ANALYSES ON AMERICAN COALS

	As received					Moisture- and ash-free				
	Moist-	V.M.,	F.C.,	Ash,	B.t.u.	H ₂ ,	C,		B.t.u.	per lb.
Mont. lignite....	24.8	35.3	31.4	8.5	6,700	1.53	5.10	70.54	1.53	21.30 12,160
Wyo. subbitumi- nous.....	20.8	35.4	40.6	3.2	10,084	0.72	6.51	59.75	0.93	31.99 13,250
Ill. bituminous low rank.....	10.8	33.1	47.4		11,524	0.87	5.92	70.75	1.53	20.93 14,350
Pa. bituminous high rank.....	3.0	22.8	64.6	9.5	13,626	3.43	5.19	84.75	1.44	5.19 15,580
W. Va. semibi- tuminous.....	2.2	18.7	74.8	4.4	14,663	1.36	4.82	87.66	1.46	4.70 15,710
Va. semianthra- cite.....	3.6	9.5	67.5	19.4	11,851	0.5				15,390
Pa. anthracite...	4.2	5.1	76.9	13.8	11,520	0.5				14,080

An important property of coals is the fusing point of the mineral matter which remains as ash when the coal is burned. The fusing point often determines the limits of application of a coal to specific burning methods, or excludes it from certain burning methods. Three types of behavior are set up for classifying coal ashes by their melting points: ashes which fuse in the temperature range from 1900 to 2200°F. are termed low fusing, those in the range from 2200 to 2600°F., medium fusing, and above 2600, nonfusing. Ordinarily, 3100°F. is the maximum melting point of ash encountered. The melting point of an ash is not predictable from the analysis of the minerals present in it, except within wide limits. The bulk of Pennsylvania bituminous coals are in the medium fusing class; anthracite in the nonfusing.

Many Midwestern coals have low fusing ashes, and give great trouble with clinkering unless special provision is made for handling them. Below are given data on ash fusibilities.

TABLE XI.—FUSIBILITY OF ASH FROM STEAM COALS OF THE UNITED STATES*

State	County	Coal bed	No. of sam- ples	Softening temp., °F.			Average analysis dry coal, per cent	
				Low- est	High- est	Aver- age	Ash	Sul- phur
Ala.	Jefferson	Pratt	7	2,290	2,580	2,430	5.49	1.59
Colo.	Huerfano	Walsen	20	2,280	2,670	2,450	11.70	0.60
Ill.	Saline	No. 5	19	1,940	2,210	2,051	9.30	3.00
	Franklin	No. 6	62	1,920	2,530	2,240	9.29	1.24
Ind.	Knox	No. 5	8	2,030	2,210	2,108	11.48	2.99
Ky.	Letcher	Elkhorn	16	2,300	2,670	2,500	3.83	0.71
	Muhlenberg	No. 9	13	1,870	2,230	1,990	9.64	3.60
Ohio.	Columbia	Middle Kittanning	5	2,454	7.35	1.81
	Jefferson	Pittsburgh	10	2,270	7.79	2.90
Pa.	Washington	Pittsburgh	22	2,010	2,610	2,327	6.75	1.46
	Somerset	Lower Kittanning	18	2,180	+3,010	+2,320	8.09	1.97
	Cambria	Upper Freeport	35	1,990	+3,010	+2,353	7.64	1.72
	Cambria	Lower Freeport	36	2,010	+3,010	+2,400	7.34	1.72
W. Va.	Marion	Pittsburgh	44	2,010	2,410	2,265	6.72	1.68
	Fayette	No. 2 gas	31	2,100	+3,010	+2,670	6.00	1.12
	Fayette	Sewell	86	2,080	2,980	2,485	3.87	0.87
	McDowell	Pocahontas No. 3	162	2,120	+3,010	+2,370	5.10	0.59
Wyo.	Lincoln	Kemmerer	7	2,170	2,310	2,250	6.73	0.63
Va.	Wise	Upper Banner	17	2,200	2,680	2,453	6.46	0.82

* From Marks' "Mechanical Engineers' Handbook," McGraw-Hill Book Company, Inc., (Bureau of Mines Bull. 209, "Fusibility of Ash from Coals of the United States," W. A. Selvig and A. C. Fieldner, 1921).

For hand-fired uses, it is desirable to burn the so-called smokeless coals. These are the low-volatile semibituminous coals of the Appalachian region. Anthracite is, of course, an excellent fuel for hand firing but is relatively expensive and therefore confined to certain districts in ordinary marketing. Underfeed stokers can very nicely use the high-volatile bituminous coals, but great care must be taken to select coals with high or medium fusing ashes in order to avoid clinker troubles. Overfeed stokers perform at their best with good grade Midwestern bituminous coal, but this stoker will burn almost any fuel. While the underfeed stoker finds its chief application in moderate-sized commercial installations, the overfeed stoker fits best into strictly

industrial uses. In larger power plants it is customary to employ the chain-grate stoker. This latter type stoker will burn the cheapest grades of fuel, and with economy. It is not bothered by low-fusing ash. Pulverized coal is as yet in the development stages; it bids to become an important method of coal burning, but numerous technical problems must still be worked out in its application.

Coke is the nonvolatile portion of coal which remains in the retort after the volatile portion of the coal has been distilled off. It consists largely of carbon, with varying amounts of ash, depending upon the ash content of the original coal. Upon exposure to air, the porous coke picks up some moisture. In typical cokes, the fixed carbon varies from 75 to 92 per cent, the volatile matter from 0.1 to 3 per cent, the ash from 5 to 20 per cent, the sulphur from 0.6 to 4.5 per cent, and the moisture from 0.2 to 2.25 per cent. The heating value of coke varies from 10,000 to 12,000 B.t.u. per lb.

By far the greater part of our manufactured coke is used in metallurgical work, especially in the iron blast furnace. Considerable quantities, however, are used as domestic fuel. Coke is rarely used as a commercial or industrial fuel under boilers, with the exception of coke breeze, a finely crushed material which results from coke manufacture and which cannot be used metallurgically or in domestic heating.

From the ultimate analysis of a coal a close approximation of the calorific value can be calculated by the Dulong equation:

B.t.u. per lb.

$$= 14,544 \times \frac{\%C}{100} + 62,100 \times \left(\frac{\%H_2}{100} - \frac{\%O_2}{800} \right) + 4050 \times \frac{\%S}{100}$$

COMBUSTION THEORIES OF COAL

Coal is a complex mixture of solid carbon and hydrocarbons and other volatile substances. During combustion, the volatile portions are driven out of the coal and the coke remains on the grates. Combustion therefore occurs in two zones: coke burns on the grates, and gases burn in the combustion space above the fuel bed. In order to point out the differences which exist between the burning of coal and the burning of oil, it will be useful to study the coal-combustion process in some detail.

Volatile matter begins to leave the coal when the temperature reaches 660°F. At 1300°F. the distillation is rapid. The hydrocarbons appear first in evolved gases; later hydrogen and carbon monoxide are formed. It must be understood that the gases which compose the volatile matter do not necessarily exist as such in the original coal. A thermal decomposition of the organic matter of the coal occurs, similar to the cracking of petroleum vapors which has been described in an earlier chapter. The

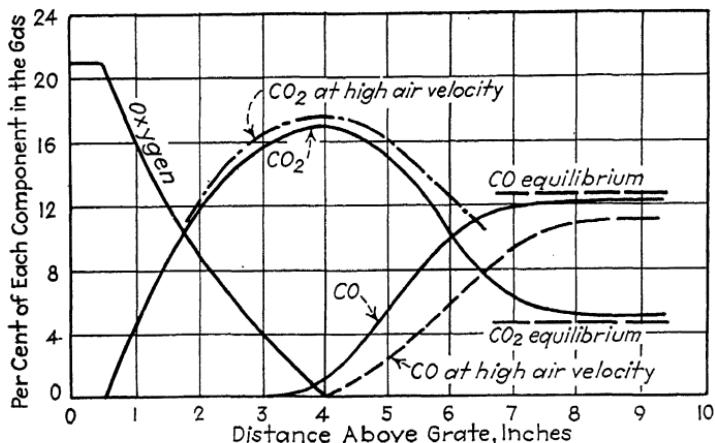


FIG. 24a.—Composition of the gases within the fuel bed of a coal-fired furnace. All the oxygen is consumed at about 4 in. above the grate. (*Haslam and Russell "Fuels and Their Combustion," McGraw-Hill Book Company, Inc.*)

volatile matter itself contains about 30 per cent of the original calorific value of the fuel, and since the coke remaining on the grates burns incompletely to form carbon monoxide which in turn must be burned in the combustion space above the fuel bed, as much as 40 to 60 per cent of the coal's total heat is released in the combustion space.

The air which enters under the grates, and ascends through the coal, is called primary air. The Bureau of Standards, among others, has conducted extensive tests on the reactions occurring between the air stream and the coal or coke as this portion of the combustion process proceeds. These tests were made to cover a great range of coal-firing conditions: the combustion rate was varied between 3 and 125 lb. of coal per sq. ft. of grate per hr., and two thicknesses of fuel bed were used, 6 and 12 in. By means of gas-sampling tubes (water cooled), located at various heights

above the grates, and of optical pyrometers for measuring temperatures, the results plotted in Fig. 24a were obtained.

In the entire range of firing rates, from 3 to 125 lb., the oxygen was always used up entirely at a height about 4 in. above the grate, or 4 in. above the ash bed if such were present on the grate, and the rate of combustion had but little effect upon the point to which oxygen could persist unchanged in the air stream. This oxygen first combines with carbon to form carbon dioxide, for it will be noted that the formation of carbon dioxide is exactly equal to the disappearance of oxygen; the two curves cross at the 10.5 per cent line, making their total 21 per cent, or the equivalent of the original oxygen of the entering air. Hence no carbon monoxide is formed in the first 4 in. above the grate. But as soon as the carbon dioxide reaches a concentration of about 16 per cent, the formation of carbon monoxide commences; this is just before the last trace of free oxygen is consumed. While free oxygen is still present, and while carbon is burning directly to carbon dioxide, the zone is termed oxidizing.

Simultaneous with the disappearance of all oxygen from the original primary air, the percentage of carbon dioxide, which has attained its maximum of about 16 per cent, begins to fall rapidly while the percentage of carbon monoxide builds up rapidly. This is brought about by the reduction of carbon dioxide by the incandescent coke of the upper fuel bed. The thicker the bed, the more carbon monoxide formed. Also, the rate of combustion, which is determined by the rate of air supply, affects the carbon monoxide formation to a greater extent than it does the formation of carbon dioxide. Since carbon monoxide is formed by the reduction of carbon dioxide in contact with glowing carbon, the longer the contact the more carbon monoxide made. Hence, while high air velocity *increases* carbon dioxide formation slightly, it *decreases* carbon monoxide formation considerably. Thick fuel beds increase carbon monoxide so that the bed should be no thicker than required by the physical maintenance of an even bed. This region is called the reducing zone.

In the concluding paragraph of the last chapter, it was pointed out that herein is a major difference between the normal combustion of coal and oil—carbon monoxide is *always* produced during the first stage of coal combustion, whereas carbon monoxide need *not* be produced *at all* during the proper burning of fuel oil.

The volatile matter first distilled off from the green coal and the additional carbon monoxide resulting from the reduction of carbon dioxide must be burned in the combustion space above the fuel bed. To accomplish this, it is necessary to introduce a supply of secondary air. From the above discussion, it is obvious that secondary air cannot be introduced through the fuel bed, as any attempt to do so would merely increase the rate of combustion on the grates, without leaving a particle of free oxygen for use above the bed. This secondary air must come into the combustion space without any previous contact with the coal.

Furthermore, it can be shown that the carbon monoxide constituent of the combustible in the gases is the most reluctant to burn completely, and that some carbon monoxide will remain even after the combustion has been allowed to proceed through an abnormally long period and when an abnormally long combustion space is provided.

The ports for admitting secondary air into the furnace are usually located in the fire door or along the fuel bed at, or just above, the level of the surface. The amount, distribution and admixture of secondary air play a heavy role in determining the efficiency of the coal combustion. Unfortunately, however, it is not easy to set up rules for amount and method of admission of secondary air for a great many factors have their influence upon this; percentage of volatile, thickness of bed, rate of combustion, available draft, desired completeness of combustion, etc. will all affect the secondary air.

The hydrocarbons which are distilled from the coal are of course similar to those present in a fuel-oil fire. Under the conditions arising from hand-fired furnace operation, it is difficult for secondary air to achieve hydroxylation of the hydrocarbon molecule, so that smoke and poor combustion are both likely to occur. Mechanical stokers help considerably in overcoming this tendency.

It has already been mentioned that there are three general types of mechanical stokers. These, together with hand firing, compose the four possible methods of burning coal. A summary of the four is given by Haslam and Russell in "Fuels and Their Combustion."

The principal methods of feeding coal and air in industrial furnaces are shown diagrammatically in Fig. 24b. In hand-fired furnaces (A)

primary air passes through a bed of ashes and comes in contact with partly burned fuel. All the oxygen is used up before the air reaches the green coal. Thus the volatile hydrocarbons distilling from the top of the fuel bed come into contact with gases depleted of oxygen, and therefore "hydroxylation" is made difficult and the deposition of soot results easily. In the overfeed type of mechanical stokers (*B*) coal from a hopper is fed onto the sloping grate and is coked and distilled in the upper part of the furnace by the heat reflected through an arch. The resulting coke burns as it travels down the grate to the ash pit. With chain-grate or traveling-grate stokers (*C*) the green coal is fed onto a

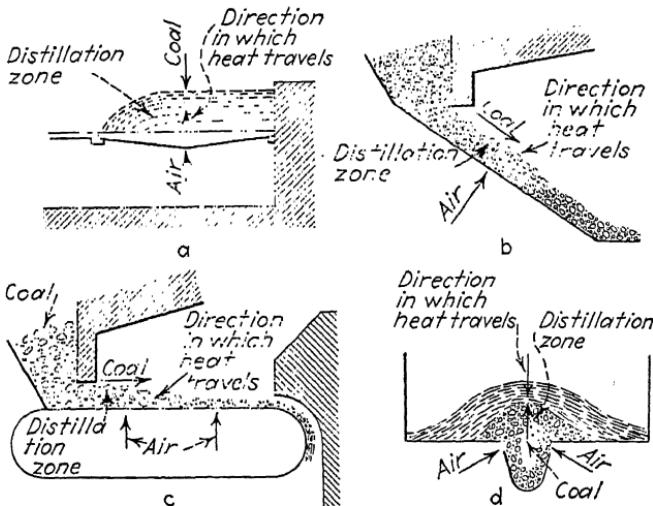


FIG. 24b. Methods of burning coal. (*Haslam and Russell, "Fuels and Their Combustion," McGraw-Hill Book Company, Inc.*)

moving horizontal grate. The coal is coked and distilled by heat reflected from an arch just as the grate enters the furnace; the resulting coke is burned during travel of the grate into the ash pit. In the under-feed type of mechanical stoker (*D*) the green coal is fed from the underneath part of the fuel and distilled volatile matter passes up through the bed of incandescent coke. Thus in overfeed stokers and in chain-grate stokers, part of the primary air, and in underfeed stokers practically all the entering primary air, passes through the green coal as it is being distilled. The distillation of volatile matter therefore takes place in an oxidizing atmosphere, the entrance of oxygen into the hydrocarbon molecule (hydroxylation) is made easier and the tendency for "cracking" or soot deposition is decreased. Moreover, with mechanical stokers, the fresh coal is heated gradually, and the volatile matter is driven off relatively slowly, again tending to promote hydroxylation.

The amount of coal fired and the amount of air supplied the furnace may be regulated automatically with practically all types of stokers; therefore it is possible to feed into the furnace the coal and air properly proportioned, and in amounts depending on the load.

DATA ON GASEOUS FUELS

While fuel gases embrace a wide variety in type, all of them are composed of but three classes of combustible, hydrogen, carbon monoxide and hydrocarbons (together with various noncombustible or inert gases such as nitrogen, water vapor and carbon dioxide). Gaseous fuels are derived from many sources, so that the various component gases that may be present in the mixture exhibit great variation in percentages between the different gas mixtures, being entirely absent in some cases and again being the predominant substance in others. Below is a brief review of the commoner gaseous fuels, showing the source or method of manufacture and chief characteristics.

Acetylene.—This is the single hydrocarbon substance C_2H_2 . It is manufactured by the action of water on calcium carbide. The principal use for acetylene, of course, is the welding and cutting torch. It is also used to some extent in small lighting plants, especially signal devices for traffic control. It is distributed and used in tanks, in which the gas is compressed up to pressures of 300 lb. per sq. in.

Blau Gas.—Petroleum vapors under high compression—up to 100 atm.

Blast-furnace Gas.—A by-product gas which issues from the top of blast furnaces used for smelting iron ore. This gas is usually used in the plant which produces it, for heating the blast stoves or air preheaters for the blast furnaces and under boilers, etc. Or it may be cleaned and used in internal-combustion engines. It is frequently mixed with producer gas. Its calorific value is low.

Blue Water Gas.—This is the initial product in the manufacture of carbureted water gas, which latter is the most important of the manufactured fuel gases. Blue water gas is made by the action of steam on incandescent coke, in which process the carbon reduces the water leaving free hydrogen and forming carbon monoxide. As the reaction absorbs a large amount of heat, the coke mass quickly cools, so that it is necessary to alter-

nately blow air and then steam through the coke bed. The coke bed thus sustains its own temperature by combustion. Blue water gas burns with a nonluminous blue fire, hence its name.

Bottled Gas.—Various hydrocarbon gases are sold, for rural use for cooking and other domestic purposes, in small tanks under compression. Propane is the most common hydrocarbon used; butane and pentane are not uncommon.

Carbureted Water Gas.—Blue water gas must be enriched with high calorific hydrocarbons to bring up its heating value to legal requirements, and also to make it luminous if it is to be used for lighting. This latter use, which years ago was its principal one, is now of less importance than its use as a fuel. The enriching consists in vaporizing gas oil under thermal conditions which make the hydrocarbons fixed gases, principally methane and ethane.

Coal Gas (Retort Gas).—Before the advent of water gas, most illuminating and fuel gas was made by the destructive distillation of coal in refractory retorts from which air was excluded. The process is still used extensively for the manufacture of coke, but the coking ovens are designed differently so as to yield the most suitable form of metallurgical coke. The volatilized matter from either type oven or retort contains tar, ammonia, aromatic hydrocarbons (principally benzene, toluene and xylenes) and water vapor. The purified gas has a high calorific value.

Coke Oven Gas.—See coal gas, above.

Hydrogen.—Pure hydrogen is seldom used as a fuel, except in such specialized applications as the oxyhydrogen torch. Hydrogen, however, is present as a component in a great many fuel gases, as is shown in the tabular information of this chapter.

Natural Gas.—Petroleum deposits in the earth are very often accompanied by pockets of gaseous hydrocarbons. These hydrocarbons are termed natural gas. Methane is the principal component gas; ethane also occurs generally, though in smaller amount, while small quantities of nitrogen usually are also present. Natural gas has the highest heat value of all fuel gases.

Natural gas is piped many hundreds of miles from the oil and gas fields to more populous districts. It may be distributed in metropolitan centers as it comes from the wells, or, as is frequently the case, it may be mixed with manufactured gas of some description.

TABLE XII.—DATA ON GASEOUS FUELS

	Hydrogen, H ₂ , %	Carbon monoxide, CO, %	Methane, CH ₄ , %	Ethane, C ₂ H ₆ , %	Briylene, C ₂ H ₄ , %	Oxygen, O ₂ , %	Other bases, G ₂	Specific gravity, air = 1	Btu, per cu. ft., gross	Btu, per cu. ft., net	Btu, per cu. ft., required for therm. gross, 100,000	Btu, per mole, gross	Source*
Blast-furnace gas.....	3.2	26.2	13.0	57.6	1.02	92	1075	92	33,387	111,290	H & R
Blue-water gas.....	51.8	43.4	10.2	3.5	1.3	0.63	310	285	323	111,290	111,290	H & R
Carbureted-water gas.....	40.5	34.0	10.2	6.1	3.0	2.9	0.5	2.8†	633	550	182	197,450	AGA
Coal gas, retort.....	47.0	9.0	34.0	6.6	1.1	2.3	0.7	0.41	634	560	150	227,606	H & R
Coal gas, coke oven.....	46.5	6.3	32.0	3.5	2.2	8.1	0.8	0.5‡	574	514	174	206,166	AGA
Producer gas, anthracite.....	17.7	23.2	1.0	8.0	1.1	0.8	0.1	0.44	700	133	61	337	AGA
Producer gas, bituminous.....	14.0	27.0	3.0	4.5	50.0	50.9	0.6	0.86	163	153	613	58,517	AGA
Natural gas, Alabama.....	97.6	0.3	2.1	0.57	986†	101	353,974	G.E.H.
Arkansas.....	99.2	0.2	0.6	0.56	1083†	100	360,077	G.E.H.
California.....	84.0	14.8	0.7	0.5	0.64	1116	1069	90	400,644	G.E.H.	
Illinois.....	37.5	59.6	0.5	1.7	1.2§	0.86	1510	1510	66	542,090	G.E.H.
Kansas.....	88.3	10.0	0.8	3.7	0.61	1046	1046	98	364,744	G.E.H.	
Kentucky.....	76.4	22.6	1.0	1.0	0.67	1171	1171	85	420,380	G.E.H.	
Louisiana.....	96.5	1.4	2.1	0.58	975	975	103	350,025	G.E.H.	
Missouri.....	84.1	6.7	0.8	8.4	0.63	958	958	104	347,512	G.E.H.	
New York.....	59.8	37.6	0.8	2.2	0.75	1268	1268	79	455,212	G.E.H.	
Ohio.....	83.5	12.5	0.4	3.8	0.66	1047	1047	96	375,873	H & R	
Oklahoma.....	75.4	17.7	0.3	6.6	0.67	1074	1074	93	385,566	G.E.H.	
Pennsylvania.....	90.0	5.0	5.0	0.60	1092	1092	100	359,718	AGA	
Texas.....	51.5	10.2	38.3	0.76	700†	700†	143	251,300	G.E.H.	
West Virginia.....	82.0	17.0	0.1	0.9	0.64	1128	1128	89	404,952	G.E.H.	

* H & R, Haslam and Russell, "Fuels and Their Combustion," McGraw-Hill Book Company, Inc.; AGA, American Gas Association, "Combustion"; GEH, "Gas Engineers Handbook," McGraw-Hill Book Company, Inc.

† Benzene, C₆H₆.

‡ Calculated calorific values.

§ Hydrogen sulphide, H₂S.

Petroleum Gas.—During the refining and transportation of petroleum and casing-head gasoline, certain low-boiling hydrocarbons evaporate to form gases which are readily condensable under pressure. These are the gases sold in steel tanks for domestic use (see Bottled Gas above). In some small communities this gas is distributed by pipe into homes. For this latter purpose, pure butane must be admixed with air to reduce its inflammability; pentane need not be.

Producer Gas.—Many industrial plants generate a fuel gas for their own use by operating gas producers. These are machines in which air and steam are forced through a bed of glowing coal, coke or other solid fuel. Because of the large amount of nitrogen introduced by the air, producer gas has a very low calorific value. The gas producer affords a means of converting low-grade solid fuels, that cannot be effectively burned by ordinary methods, into a more convenient fuel which can be utilized to better advantage and at higher efficiency.

Pintsch Gas.—A gas, formerly made by cracking petroleum vapors by heating them in a closed iron vessel, was called Pintsch gas. It was extensively used, compressed in cylinders, for railway lighting. Its use is practically discontinued.

THE ATMOSPHERIC GAS BURNER

Most atmospheric gas burners devoted to domestic heating and small commercial work are constructed to embody the following elements.

1. An orifice, which has two important functions:
 - a. To regulate the rate of gas feed to the burner. For any given orifice, the gas flow is determined by the gas pressure in the service pipe. When the orifice on a burner is fixed, the gas pressure is variable by a governor or pressure regulator provided with the burner. If the orifice is variable, no pressure regulator is needed.
 - b. To create a condition in the gas stream which will cause inspiration of primary air through a port, so that gas and primary air may be mixed before reaching the place of combustion.
2. A primary air port, with suitable shutters or other adjusting device.

3. A mixing tube, in which the gas and inspirated primary air can be thoroughly mixed.
4. A burner head, which consists of an arrangement at the end of the mixing tube, with ports or slots through which the gas-air mixture can issue and begin to burn. It is important that its design insure sufficient velocity to maintain combustion outside the ports or slots and prevent the mixture in the mixing tube from burning, and at the same time hold

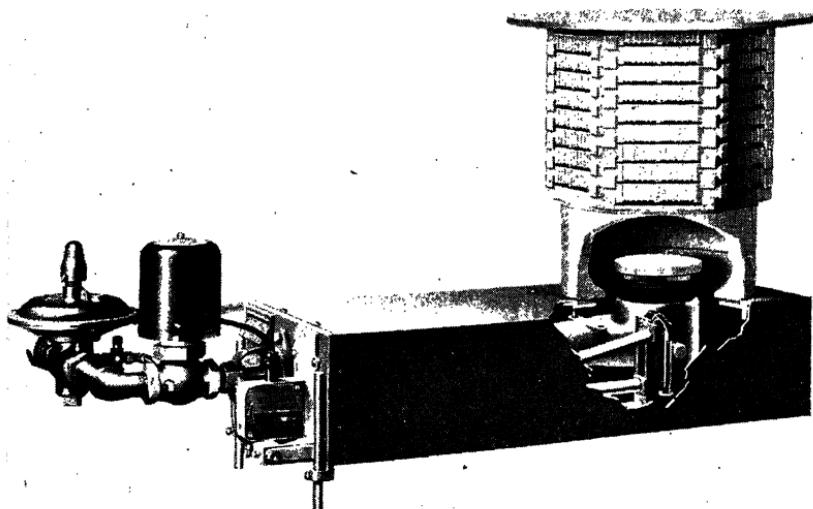


FIG. 25.—Typical atmospheric gas burner for domestic heating. (*Courtesy of Republic Flow Meters Company.*)

the flame close to the ports, so that combustion will not occur above the head.

5. A supply of secondary air, to furnish oxygen with which combustion can be completed. Usually some sort of duct system is incorporated so that required air can be taken through in regulated amount, and so that all other air is excluded, *i.e.*, so that no air enters the combustion space except as primary air, or as secondary air through the proper duct or other means provided for this.
6. Refractories or baffles, the purposes of which are:
 - a. To deflect the flames or hot gases of combustion over to the walls of the firebox in order to secure a scrubbing

action against the walls and thereby increase the amount of heat released before the gases enter the boiler flues or furnace passages.

b. To increase the amount of heat transferred to the boiler or furnace by radiation, which follows from the fact that

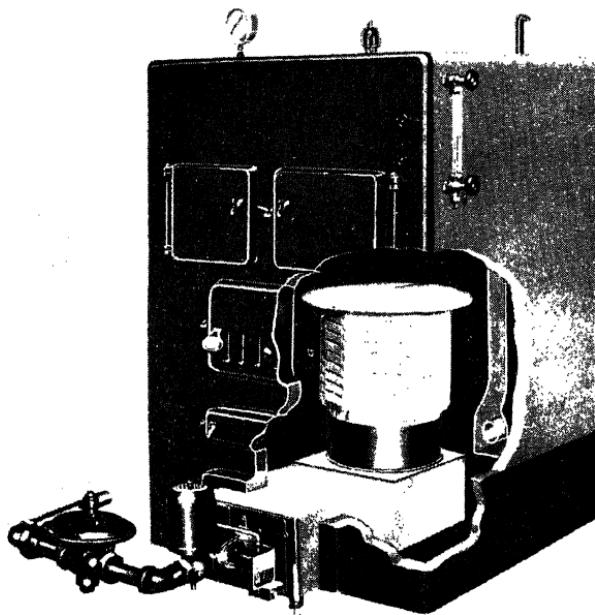


FIG. 26.—Typical atmospheric gas burner in domestic heating boiler. (*Courtesy of Republic Flow Meters Company.*)

the refractory heats up to incandescence and then radiates heat at a high rate to the walls and top of the firebox.

THE FORCED-DRAFT GAS BURNER

Forced-draft gas burners can be and are usually much simpler in design than atmospheric types. Very often they are essentially only mixing rings or chambers through which the gas passes before being forced out through ports or slots into the firebox, the mixing compartment being under pressure from an air blower. Generally, these burners are arranged for horizontal firing, and very often they are incorporated as part of combination oil and

gas burners or pulverized coal and gas burners. More discussion of this type of equipment will be found in Chap. XV.

ECONOMICS OF THE THREE FUELS

In handling combustion equipment in the field, with any of the three principal fuels—coal, oil or gas—practical limitations often

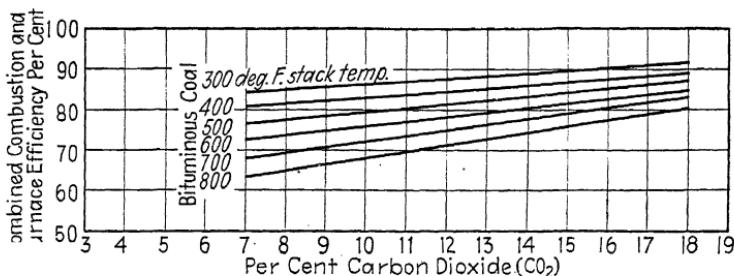


FIG. 27.—Loss of heat in stack gases for bituminous coal.

restrict the amount and extent of the test work which can be performed to determine actual operating performance and efficiency. The most general means available is the Orsat analyzer, in several different forms, for measuring percentage of carbon dioxide in the flue gas. By referring back to Chap. III it can be seen that a

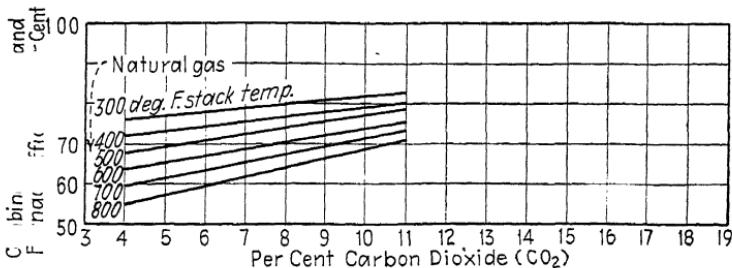


FIG. 28.—Loss of heat in stack gases for natural gas.

simple carbon dioxide reading does not tell a complete story insofar as combustion efficiency is concerned, for the maximum attainable carbon dioxide is dependent upon the amount of hydrogen in the combustible—since hydrogen in burning forms water which condenses out of the flue gas in the analyzer. Hence it is also necessary to determine oxygen as well, unless the percentage of hydrogen in the fuel is already known. Finally, since carbon monoxide is always formed in the combustion of coal or coke, and since all the carbon monoxide cannot be burned up

in the secondary combustion of a coal or coke fire, it is essential to know the extent of completeness of combustion with these fuels. This applies also to gas when carbon monoxide is one of the components of the fuel, as is usually the case. But the correct determination of carbon monoxide in flue gas is a matter of some analytical difficulty. Hence correct conclusions as to operating efficiency cannot always be drawn from the ordinary flue-gas analysis.

The accompanying charts show combustion efficiencies for various fuels, but it must be distinctly remembered that these charts show results only for the data shown, and that deviations may mean considerable changes in efficiency.

The most dependable method of ascertaining efficiency is to perform an evaporation test, if the fuel is being burned under a boiler, or to determine in some dependable way the actual amount of heat converted from the fuel into useful purposes.

The translation of quantity from one fuel to another involves two factors: the unit heat value of each fuel and the efficiency of utilization of each fuel. While the first of these is a readily determinable figure for most common fuels, the second may be somewhat involved. Efficiency of utilization implies not only actual combustion efficiency, but also regulation of combustion rate, accuracy of control or proportioning of heat release to load requirements, etc., aside from any consideration of relative labor or fixed costs. The quickest and simplest way of making comparisons of relative costs of fuels for equivalent results is to express available heat units purchasable per dollar with allowance for the utilization factor.

Let

$$H_o = \text{B.t.u. in one gallon of oil}$$

$$P_o = \text{price of oil in cents per gallon}$$

$$U_o = \text{efficiency of utilization of oil}$$

$$\text{then Available B.t.u. purchased per dollar} = \frac{100}{P_o} \times H_o \times U_o$$

$$H_c = \text{B.t.u. in one ton of coal}$$

$$P_c = \text{price of coal in cents per ton}$$

$$U_c = \text{efficiency of utilization of coal}$$

$$\text{Available B.t.u. purchased per dollar} = \frac{100}{P} \times H_c \times U_c$$

H_g = B.t.u. in 1000 cu. ft. of gas

P_g = price of gas in cents per 1000 cu. ft.

U_g = efficiency of utilization of gas

$$\text{Available B.t.u. purchased per dollar} = \frac{100}{P_g} \times H_g \times U_g$$

or if

P_t = price of gas in cents per therm

$$\text{Available B.t.u. purchased per dollar} = \frac{100}{P_t} \times 100,000 \times U_g$$

DATA ON HEATS OF COMBUSTION OF HYDROCARBONS

Since the combustible substance in all fuels is composed entirely of carbon and hydrogen (except for sulphur, which may be present in small amount as an impurity), it might at first glance be supposed that the calorific value of a fuel could be calculated from the percentage composition of carbon and hydrogen. The calorific values of pure carbon and pure hydrogen are of course known. These values are: for carbon, 174,000 B.t.u. per mole or 14,500 B.t.u. per lb.; for hydrogen, 123,100 B.t.u. per mole or 61,060 B.t.u. per lb. But when carbon and hydrogen enter into chemical combination certain heat quantities are involved in the chemical union. In some cases, heat is released when the elements unite to form compounds, and the compounds are said to have exothermic heats of formation. In other cases, heat is absorbed by the uniting elements forming the compound, and the heats of formation are then called endothermic. When either type of compound burns the heat of combustion will be influenced by the heat of formation of the compound.

According to a fundamental law of chemical thermodynamics, the amount of heat released in the change of any elements into any ultimate compounds is the same regardless of the nature or number of intermediate occurrences. Thus, when carbon and hydrogen first unite to form an hydrocarbon which then burns to form carbon dioxide and water, the total heat release for the series of reactions is identical with that occurring when the elementary carbon and hydrogen burn directly to carbon dioxide and water. It follows that when a heat of formation is exothermic, the heat of combustion of the compound is less than the heat of combustion of the original elements, and that when the heat of

TABLE XIII.—EXPERIMENTALLY DETERMINED HEATS OF COMBUSTION OF HYDROCARBONS*

Saturated Aliphatic Hydrocarbons (Paraffin Series)

Form- ula	Name	Molec- ular weight	Kg.- cal. per g.-mole	B.t.u. per lb.-mole	B.t.u. per lb.
CH_4	Methane (<i>g</i>)	16	210.8	379,440	23,715
C_2H_6	Ethane (<i>g</i>)	30	368.4	663,120	22,104
C_3H_8	Propane (<i>g</i>)	44	526.3	947,340	21,530
C_4H_{10}	Isobutane (<i>g</i>) (trimethyl- methane)	58	683.4	1,230,120	21,209
C_5H_{12}	<i>n</i> -Pentane (<i>g</i>)	72.10	838.3	1,508,940	20,930
	<i>n</i> -Hexane	86.11	991.4	1,784,520	20,724
C_7H_{16}	<i>n</i> -Heptane	100.13	1,137.3	2,047,140	20,444
C_8H_{18}	<i>n</i> -Octane	114.14	1,302.7	2,344,860	20,544
$\text{C}_{10}\text{H}_{22}$	Decane	142.18	1,610.2	2,898,360	20,315
$\text{C}_{16}\text{H}_{34}$	Hexadecane (<i>s</i>)	226.27	2,559.1	4,606,380	20,360
$\text{C}_{20}\text{H}_{42}$	Eicosane (<i>s</i>)	282.34	3,183.3	5,729,580	20,293

Naphthenes, (Polymethylenes)

C_3H_6	Trimethylene (<i>g</i>)	42	496.8	894,240	21,290
C_5H_{10}	Methylcyclobutane	70.08	784.2	411,560	20,142
C_6H_{12}	Cyclohexane	84.10	936.3	,685,340	20,040
C_7H_{14}	Cycloheptane	98.11	,087.3	,957,140	19,948
C_8H_{16}	1,2,4-Trimethylcyclopentane	112.13	,245.4	,241,720	19,992
C_9H_{18}	Methyl-1- <i>n</i> -propylecyclopentane	126.14	,401.4	,522,520	19,997
$\text{C}_{10}\text{H}_{18}$	Fenchane	138.14	,502.8	2,705,040	19,582
$\text{C}_{14}\text{H}_{26}$	3,3-Dimethyldicyclohexyl	194.21	,105.9	3,790,620	19,518

Unsaturated Aliphatics, (Olefin Series)

C_2H_4	Ethylene (<i>g</i>)	28	331	596,880	21,317
C_3H_6	Propylene (<i>g</i>)	42	490	882,360	21,009
C_4H_8	Isobutylene (<i>g</i>)	56	647.	1,164,960	20,802
C_5H_{10}	Amylene	70	803.	1,446,120	20,659
	Hexylene	84.10	952.6	1,714,680	20,390
	Diisobutylene	112.13	252.4	2,254,320	20,105
$\text{C}_{10}\text{H}_{20}$	Diamylene	140.16	582	2/2,847,960	20,320
$\text{C}_{12}\text{H}_{24}$	Triisobutylene	168.19	1,858	3/3,344,940	19,888

Aromatic Hydrocarbons

C_6H_6	Benzene (<i>v</i>)	78.05	787.2	1,416,960	18,154
C_6H_6	Benzene	78.05	783.4	1,410,120	18,064
C_7H_8	Toluene	92.06	937.0	1,686,600	18,321
C_8H_{10}	<i>o</i> -Xylene	106.08	1,094.2	1,969,560	18,566
C_9H_{12}	Isopropylbenzene	120.10	1,247.3	2,245,140	18,694
	1,3-Isopropyltoluene	134.11	1,409.5	2,537,100	18,918
	Pentamethylbenzene	148.13	1,554.0	2,797,200	18,883

TABLE XIII.—EXPERIMENTALLY DETERMINED HEATS OF COMBUSTION OF HYDROCARBONS*.—(Continued)

Form- ula	Name	Molec- ular weight	Kg.- cal. per g.-mole	B.t.u. per lb.-mole	B.t.u. per lb.
Aromatic Hydrocarbons.—(Continued)					
C ₁₂ H ₁₈	Hexamethylbenzene (s)	162.14	1,711.9	3,081,420	19,005
C ₁₃ H ₁₂	Diphenylmethane (s)	168.10	1,655.0	2,979,000	17,724
C ₁₄ H ₁₄	Dibenzyl (s)	182.11	1,810.6	3,259,080	17,896
C ₁₉ H ₁₅	Triphenylmethyl (s)	243.12	2,378.5	4,281,300	17,700
C ₂₅ H ₂₀	Tetraphenylmethane (s)	320.16	3,102.4	5,584,320	17,442
Naphthalene (s)					
C ₁₂ H ₁₀	Diphenyl (s)	128.06	1,232.	2,218,500	17,324
C ₁₃ H ₁₀	Fluorene (s)	154.08	1,493.6	2,688,480	17,450
C ₁₄ H ₁₀	Anthracene (s)	166.08	1,584.9	2,852,820	17,742
C ₁₈ H ₁₈	Chrysene (s)	178.08	1,693.4	3,048,120	17,116
C ₁₈ H ₁₈	Methylisopropylphenanthrene (s)	234.14	2,306.8	4,152,240	17,734
C ₂₄ H ₁₈	1,3,5-Triphenylbenzene (s)	306.14	2,936.7	5,286,060	17,270
C ₂₈ H ₂₀	Dianthracene (s)	356.16	3,382.9	6,089,220	17,100
Unsaturated Aromatics					
C ₈ H ₈	Styrene (phenylethylene)	104.06	1,045.4	1,881,720	18,083
C ₉ H ₁₀	α -Methylstyrene	118.08	1,202.9	2,165,220	18,337
C ₁₀ H ₁₂	β -Ethylstyrene	132.10	1,346	2,422,980	18,342
C ₁₁ H ₁₄	Phenyl-1-pentene-2	146.11	1,510.0	2,718,000	18,602
C ₁₂ H ₁₆	β,β -Diethylstyrene	160.13	1,664.9	2,996,820	18,714
C ₁₄ H ₁₂	Stillbene (s)	180.10	1,765.0	3,177,000	17,640
C ₁₅ H ₁₄	α,β -Methylphenylstyrene (s)	194.11	1,929.2	3,471,560	17,884
C ₁₆ H ₁₄	Diphenylbutadiene (trans-trans)	206.11	2,030.3	3,654,540	17,731
C ₁₈ H ₁₈	Dibenzylbutadiene (s)	234.14	2,341.0	4,213,800	17,997
C ₁₈ H ₂₀	Diphenyl-1,4-ethyl-1-butene- 3	236.15	2,372.6	4,270,680	18,085
C ₂₀ H ₁₆	Diphenylstyrene (s)	256.13	2,508.6	4,515,480	17,629
Hydroaromatic Hydrocarbons					
Dihydrobenzene					
C ₇ H ₁₂	Dimethylmethylenecyclopro- pane	80.06	833.2	1,499,760	18,733
C ₈ H ₁₂	Cycloheptene	82.08	898.0	1,616,400	19,693
C ₈ H ₁₂	1,3-Dimethyldihydrobenzene	108.10	1,148.2	2,066,760	19,110
C ₉ H ₁₄	Ethyl-1-cyclohexene-1	110.11	1,203.7	2,166,660	19,677
C ₁₀ H ₁₀	1-Methyl-4-ethylcyclo- hexadiene-1,3	122.11	1,310.8	2,359,440	19,322
C ₁₀ H ₁₀	Δ_1 -Dihydronaphthalene	130.08	1,296.	3,333,340	17,938
C ₁₀ H ₁₄	Hexahydronaphthalene	134.11	1,419.	5,554,740	19,049
	Menthene	138.14	1,523.	7,741,760	19,848

TABLE XIII.—EXPERIMENTALLY DETERMINED HEATS OF COMBUSTION OF HYDROCARBONS.*—(Continued)

Form- ula	Name	Molec- ular weight	Kg.- cal. per g.-mole	B.t.u. per lb.-mole	B.t.u. per lb.
Acetylene Hydrocarbons					
C ₂ H ₂	Acetylene (g)	26.02	312	561,600	21,583
C ₃ H ₄	Alllylene (g)	40.00	465	837,180	20,929
C ₆ H ₆	1,5-Hexadiine	78.05	853	1,536,300	19,683
C ₇ H ₁₂	Heptine-1	96.10	1,091	1,964,160	20,439
	Phenylacetylene	102.05	1,024	1,843,560	18,065
C ₁₀ H ₁₀	Phenyl-1-butine-3	130.08	1,340	2,612,000	20,080
	Diphenyldiacetylene	202.08	1,975	6,3,556,080	17,957

* Based upon initial and final temperatures at room temperature. *g* indicates gas; *s*, solid; *v*, vapor.

All other values are for liquids.

formation is endothermic, the compound will have a higher heat of combustion than would the original elements in burning directly.

In even the most careful measurements of heats of combustion, compared to theoretical calorific values, there is some deviation. The data in Table XIII show the latest available calorific values of some hydrocarbons. For a longer list of hydrocarbons and other types of chemical compounds, the reader is referred to the source from which those data were taken, "Heats of Combustion of Organic Compounds" by M. S. Kharasch, published by the Bureau of Standards as *Research Papers* 41, 1929.

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CHAPTER V

PROBLEMS OF OIL-BURNER DESIGN

The term oil burner is applied to any device which has the function of doing one or more of the following: preparing oil for combustion by changing the liquid into a spray, emulsion or vapor (in other words to cause the liquid to simulate the gaseous phase); mixing this quasi-gas or vapor with air in order that combustion may be initiated; starting the combustion by means of an ignition system (in automatic operation); causing the combustion to be sustained by regular feeding of the oil and admixture with air.

It happens, therefore, that the oil burner rarely is responsible for complete combustion of the fuel, except in the sense of initiating it under suitable environment. The environment itself must induce and permit of completion of combustion. Sometimes it is difficult to distinguish just where the burner and the environment merge. For instance, secondary air may be introduced in a variety of ways, and in some cases the secondary air port is an integral part of the burner, while in other cases it distinctly is not. Hence the term oil burner is somewhat of a misnomer, but the word has become well established in the vernacular of engineering so that little confusion results from its use.

Just as fire is one of those servants of mankind which was put to practical use long before its scientific nature was known, so oil burners were made and oil used as a fuel many years before the exact physics and chemistry of oil-combustion and petroleum science were worked out. It is but natural therefore that oil-burner design began as an art rather than a precise branch of engineering as it is today. Countless individuals have contributed their efforts in bringing the practice of oil burning to its present state of development. To these individuals great credit is due; but it is now possible to set forth the science of oil-burner design in more or less concise fashion and to abandon much which has been considered as sound practice heretofore. It is still true, of course, that a great deal remains to be established,

for in this, as in all branches of engineering, organized research will bear fruit. But the industry has progressed so far, and the sum total of the now available knowledge is so vast, that it is futile to expect that mere puttering with oils and burners will develop anything new or profound. There is need for continued endeavor towards perfection, but this must be carried on in well-equipped laboratories and shops under men well trained in basic engineering and experienced in present practice.

The first and principal requirement of an oil burner is that it change the physical state of the fuel oil from a liquid to a gas or to a condition resembling a gas. It will be recalled from the previous chapters that combustion consists of the chemical union of substances with oxygen, and that in the case of hydrocarbons it is essential to proper combustion that the oxygen atom enter the hydrocarbon molecule before cracking occurs. In so entering the hydrocarbon molecule, oxygen hydroxylates the fuel and makes possible the immediate formation of carbon dioxide and water without the intermediate formation of carbon monoxide. Such introduction of oxygen into the hydrocarbons cannot be accomplished while it remains a liquid. So the first problem in burning oil is to secure the intimate mixture with air which will permit hydroxylation to occur, and this problem can only be satisfactorily met by changing the fuel from a liquid to a gas or near gas.

To this process of altering the physical state of the fuel to one of attenuation the general term of atomization is applied. There are many methods of atomization used in commercial practice, but they all have a common end, *viz.*, to subdivide the fuel into as small particles as possible and to mix those particles with air as thoroughly as possible. In a true gas or vapor, of course, the substance actually exists as discrete molecules, which is to say that the particles are submicroscopic in size. But a substance can exist as a gas only when its temperature is above its boiling point at the particular pressure obtaining (usually atmospheric). Hence a burner converts the fuel into a true gas only in those few cases when heat is used to vaporize the liquid. These so-called vaporizing burners were once extensively used in domestic work, but are now less commonly encountered. All other atomizing methods convert fuel oil into a spray, where the smallest particle is vastly larger than molecular dimensions.

Nevertheless, the actual surface of fluid exposed to contact with air is increased several thousand-fold by this change from liquid into spray, and since the reaction between hydrocarbon and air is entirely a function of the surface of contact, the desired end is attained. It follows that, in general, as between various methods employed for atomization, the finer the spray the better the method, and the better the method the greater probable resultant efficiency of utilization of the fuel.

Of course, a given atomizing method will not produce the same fineness of spray with all fuels. Some methods are restricted entirely to lighter domestic oils. Others will successfully break up heavy oils up to a maximum viscosity; to use this method on heavier oils necessitates that the oil be preheated till its viscosity be reduced to that maximum which the atomizing method can handle. This matter of viscosity, it will be developed, is the most important characteristic of a fuel oil insofar as its application is concerned. Later chapters of this book, in treating of different types of oil burners, will stress this interrelationship of viscosity and utility.

Viscosity is largely a characteristic of the internal constitution or molecular structure of a substance. In this respect, it may be difficult for laboratory research to reveal much of additional importance on the matter of atomization. But viscosity is also related to surface tension, and it seems more probable that investigation of this phenomenon with respect to its bearing on atomization will be the more fruitful. It is the opinion of the author that the resources of the modern physical or physicochemical laboratory should be brought to play on the study of surface tension in an effort to develop better methods of oil atomization for oil burners.

The exact amount of air used for hydroxylation, or as primary air, is determined largely by the method of atomization employed and the character of the fuel oil burned. The total amount of air required for complete combustion, on the other hand, is entirely a matter of the percentages of carbon and hydrogen in the fuel, and this ratio varies only within close limits. It follows, therefore, that the amount of primary air to be used in any given case must be studied out for practically every type of atomization. This will be amplified in the individual chapters on burner types that follow.

Once the burner has started combustion, the environment generally sustains it as long as the burner continues to supply oil properly atomized and mixed. Sometimes the burner supplies the secondary air; more often the environment supplies it. The burner must do one thing; it must supply fuel, either continuously but at a varying rate or intermittently but at a constant rate, in a manner conforming fairly closely to the load demand. As those cases are rare indeed where the load is continuous and unvarying, the burner must be capable of adjustment to load fluctuations or able to stop and start sufficiently often to meet this requirement. There are in use a great many different types of control methods, which exhibit great diversity in their nature. Sometimes automatic control is entirely lacking, and the burner is of the so-called straight manual type. Sometimes the automatic-control system is of considerable complexity, embracing regulation of a host of different factors and causing the burner to respond to changes of such variables as load, boiler pressure, air temperature, outdoor temperature, time, draft intensity, water temperature, water level, etc. Oil-burner control in general requires regulation of oil flow, air flow, atomization, mixing, efficiency of combustion; interlocked with these are problems of oil feeding and air introduction. Just how all of these have been or are to be worked out for all applications are the subject matters of the following chapters.

UNDERWRITERS' LABORATORIES' STANDARDS FOR OIL BURNERS

Underwriters' Laboratories of Chicago publish a bulletin covering their requirements for domestic fuel-oil burners. Their designation "domestic" signifies an automatic oil burner, regardless of the design, construction, application or grade of oil used. They consider oil burners from the following points of view:

- a. Simplicity of construction.
- b. Rigidity of assembly to withstand effects of external shock.
- c. Arrangement to minimize possibility of tampering by unauthorized persons.
- d. Shielding of rotating parts and electrical parts and connections.
- e. Durability of the parts.
- f. Safeguarding against abnormal discharge of fuel oil.

- g. Reliability of operation of the burner and safety device.
- h. Reliability of ignition system.

Adequacy of manufacturer's instructions on installation, operation and maintenance.

- j. Manufacturers' facilities for manufacturing.

Some of the more important points on which they set up standards for operation are:

In the case of multiple-burner installations in a single heating unit, the automatic safety-control method of each burner shall operate independently of the other, or equivalent safety features shall be provided so that in no case can one burner operate unsafely.

Burners shall be uniform and reliable in operation, as odorless as practicable, and free from hazardous carbonization or any other phenomena which may lead to tampering by the user. The design should be such that the possibility of explosions and the generation or accumulation of unburned vapors or gases are reduced to a minimum.

Burner shall be capable of operating acceptably at any voltage between 85 and 110 per cent of rated voltage for alternating current and between 80 and 110 per cent of rated voltage for direct current, both during normal conditions and when conditions of low oil temperature (32°F.) are imposed in tests.

The burner design shall be such as to insure a uniform and supply of fuel and air for combustion. The means for regulating supply of air and oil shall be arranged so that the adjustment may be fixed or restricted in a manner to discourage tampering. The burner design shall be such that when the means for regulating the air supply is adjusted to its minimum, sufficient air will be admitted to the burner to sustain combustion. Burners depending entirely upon natural draft for their air supply shall be provided with a draft regulator which is capable of maintaining the draft for which the burner is designed.

Burners of the swinging type shall be provided with reliable means of locking the burner in the firing position and for preventing starting or operation when in other than the firing position.

Burner parts intended for the handling of fluids under pressure shall be designed for six times the maximum working pressure.

Burners shall be arranged so that they may be installed in accordance with the National Electrical Code, as regards their electrical equipment.

Low-voltage circuits are defined as currents of 50 volts or less from batteries or 25 volts or less from transformers; high-voltage circuits are defined as having voltages under 600 and over those defined as low voltage; high-tension circuits are those in which the voltage exceeds 600.

Low-voltage devices and assemblies shall be capable of withstanding a dielectric-strength test of 50 volts imposed for 1 min.; high-voltage

devices and assemblies (except motors) shall be capable of withstanding a dielectric-strength test of two times the rated voltage plus 1000 volts for 1 min.; capacitors shall have the same dielectric strength as high-voltage devices.

Gauge glasses or sight feeds in oil-burner devices, the breakage of which allows a discharge of fuel oil from the fuel-oil system, shall not be used.

The use of pressure gauges as regular equipment is not recommended, but where used the scale range of the gauge shall be at least twice the maximum normal operating pressure and greater than the maximum pressure obtainable with the burner.

Bare high-tension conductors shall be solid. Bare electrode, bus bars and terminals shall be spaced away from all other metal parts and be so insulated and arranged as not to arc-over at any point throughout the assembly when a voltage 50 per cent in excess of the maximum possible voltage to ground is impressed for a period of 1 min. between the electrode assembly and ground.

Electrode tips shall be of such design and material that extreme burning of the points will not result after a reasonable period of use. Suitable high temperature alloy steels or equivalent material shall be used for electrode tips.

Insulators shall provide a distance, as measured across the surface of the insulators, between the nearest point of bare current-carrying parts and the nearest electrically grounded metal surface as indicated below.

Secondary Voltage of Ignition Transformer	Minimum Surface Distance over Insulator, In.
Not more than 5,000	1
Not more than 10,000	1½
Not more than 15,000	2

It is recommended that the oil line from the pressure regulating valve to the nozzle of pressure atomizing burners be not greater than $\frac{1}{8}$ in. inside diameter to overcome trapping of air.

The base or housing of an oil pump shall be designed to provide means for mounting it independently of the liquid-handling pipe and oil storage tank.

Stuffing boxes of conventional design shall be provided with a removable, shouldered, unthreaded follower gland. The assembly shall be designed with a nut, yoke or other acceptable means for adjusting the gland to exert pressure on the packing as wear occurs. The gland shall be beveled at the lower end, and a corresponding angle of bevel shall be provided in the shoulder at the bottom of the stuffing-box recess.

Pumps shall be capable of maintaining not less than a 25 in. vacuum when operated with an unrestricted discharge and shall be capable

of maintaining a vacuum of not less than 10 in. at its normal discharge pressure and volume.

Safety combustion controls shall stop the operation of the burner within a predetermined safe period of time in case of ignition failure or premature extinguishment of flame. Such controls shall allow burner operation only when the safety-control circuit is closed. The safety switch shall be operated independently of the current consumed by the burner and shall operate to shut off the burner in not more than 2 min. upon ignition failure or premature flame failure.

Small orifices liable to become clogged shall be protected by a strainer. The largest screen opening of the strainer shall have its smaller dimension not greater than 75 per cent of the smaller dimension of the smallest port protected by the screen. The effective screen area of strainers for capacity not more than 10 g.p.h. shall provide a total area of screen openings per gallon of fuel per hour as follows: For Nos. 1 and 2 fuel oil, at least $\frac{1}{2}$ sq. in., for Nos. 3 and 4 fuel oil, at least 1 sq. in. and for Nos. 5 and 6 at least 1 sq. in., but not less than 10 sq. in. total area of screen openings.

When the capacity exceeds 10 g.p.h., the effective screen area shall provide a total area of openings of 10 sq. in. plus $\frac{1}{2}$ sq. in. for each gallon per hour in excess of 10 g.p.h.

Secondary strainners shall be employed to protect all orifices smaller than $\frac{1}{32}$ in. diameter, should there be any iron pipe or other fuel conduit beyond the primary strainer likely to put scale into the fuel, and shall be incorporated into the burner assembly if a primary strainer is not a part thereof.

Primary strainners shall be free from external leakage when subjected to a hydrostatic pressure of 60 lb. per sq. in. The screen with its openings clogged against flow shall withstand, without collapsing or breaking, a pressure on the inlet side of 18 lb. (28 in. of vacuum plus 8 ft. head pressure). Primary strainners in burners employing a pressure pump shall be installed on the suction side of the pump. Secondary strainners installed on the pressure side of pumps shall be designed so that the housing will withstand without leakage or damage a hydrostatic pressure of three times the maximum possible pressure of the burner with which it is to be employed. Strainners shall be marked with an arrow or other symbol to indicate the correct direction of oil flow.

Manually operated oil-regulating valves employed in the assembly of oil burners shall be designed with limits which may be permanently set to restrict the maximum amount of oil delivered to an amount which can be properly consumed, or the valve shall be properly enclosed or shielded to discourage tampering after adjustment has been made by the installer.

CHAPTER VI

DOMESTIC OIL BURNERS

In this book, oil burners are classified under three types of applications: domestic, commercial and industrial. There are fairly well-defined limiting characteristics to these three groups of burners, though, of course, in several respects there is an overlapping of both principles and applications. Distinctions between the three groups must be drawn on the bases of grades of oil handled, construction and design, and capacity.

Domestic burners themselves vary so much in design and construction, and in atomizing methods especially, that it has become necessary to have available for them at least four grades of fuel oil which are commercially recognized, while occasionally a burner manufacturer calls for some special features in an oil for his burner so as to have it work at maximum effectiveness. Methods of atomization employ a variety of physical and chemical means; manufacturers using similar atomizing methods will vary the form and arrangement of burner parts and controls; even in respect to limits of capacity, there is no uniformity of opinion, for some manufacturers produce sizes or models of domestic burners having capacities beyond any reasonable maximum. That is why it is necessary, in order to distinguish the qualifying characteristics of this group, or any other of the oil-burner groups, to apply all three tests: grade of oil burned, construction and design, and capacity. These will be the chief topics discussed as various types of burners are considered.

Domestic burners may be conveniently subdivided according to methods of atomization. At least five distinct or fundamental methods exist:

1. Vaporization by heat.
2. Atomization by low oil and air pressure.
3. Atomization by high pressure and centrifugal action.
4. Atomization by centrifugal force and air stream, from
 - a. Vertical rotating cup.
 - b. Horizontal rotating cup.
5. Air atomization.

VAPORIZATION BY HEAT

The earliest successful domestic burners were of the heat-vaporizing type, starting about 1918 or 1920. Then, about 1925, other types rapidly displaced the original types of heat vaporizers from the market, while a different type of heat vaporizer also became popular. Now, in 1936, burners similar to the first vaporizers, but modified sufficiently to overcome some of the early difficulties, are again enjoying a degree of public acceptance. These burners have always had in their favor the advantages of low manufacturing cost and mechanical simplicity.

Figures 29 and 30 illustrate two forms of heat vaporizers that were popular and enjoyed wide acceptances about 1920. In the

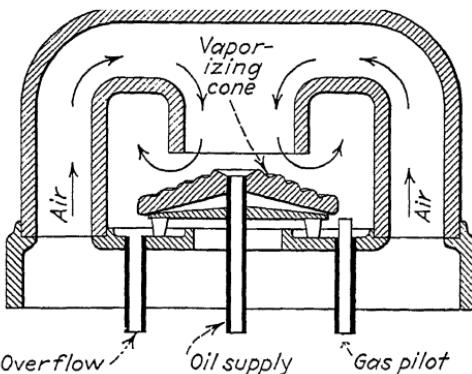


FIG. 29.—Early style of vaporizing burner, natural draft.

first form oil was spread out as a thin film over the surface of a terraced cone and so subjected to evaporation from the temperature of the metal cone. This cone was maintained at the vaporizing temperature either through a continuously burning oil flame or by means of a gas flame. In either case, if the burner was controlled automatically, the oil or gas flame also served to ignite the main oil stream whenever the control system called for heat. Almost the entire air supply was delivered through the manifold above the vaporizing plate, where it was subjected to some preheating before coming into contact with the fuel. The air supply was sometimes induced purely by natural draft; in other cases a fan was employed to force air through. In the second form (Fig. 30) the hot plate used for vaporization was contained within a combustion pot, constructed of alloy steel

with heat-resistant properties, often being lined with refractory material. The air supply was generally furnished from a blower

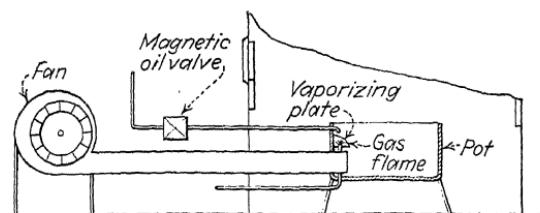


FIG. 30.—Early style of vaporizing burner, with fan.

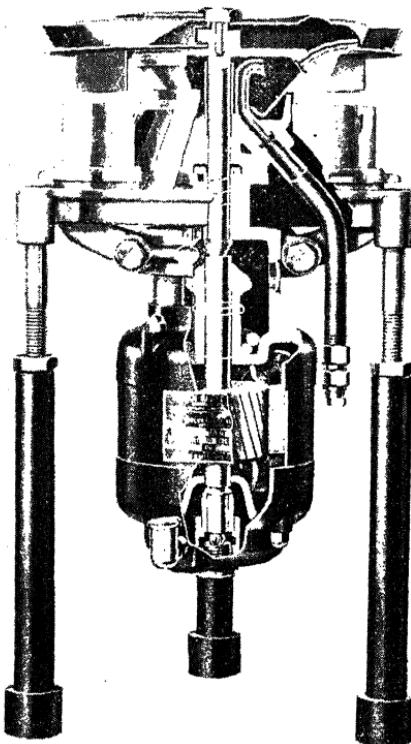


FIG. 31.—Vertical spinner burner, vaporizing type, giving wall-wiping blue flame. (Courtesy of Timken-Detroit Axle Company.)

which delivered air through a tube that entered the pot at a point most strategic for securing good admixture with the gasified oil.

Both of the above designs are typical of a large number of similar burners which were produced during the early days, especially as regards the simple mechanism exemplified in Fig. 29. But there were certain inherent faults in these heat vaporizers. Of these, the most serious was the coincidence of the end point of the fuel with the minimum cracking temperature of the hydrocarbons composing it. Thus the maximum permissible end point of a No. 1 oil is 600°F. But at that temperature some thermal decomposition of the fuel commences, hence making the formation of carbon almost inevitable even with the ordinary grades of No. 1 oil. The use of heavier oils or of inferior grades of No. 1 is precluded, thereby restricting these burners to the highest priced fuel. Another objectionable feature is the relative expense of maintaining the gas pilot constantly burning. Finally, because of the delicacy of adjustment required in setting the air-oil rate, smoking occurs rather oftener than is desirable, and regulation of combustion efficiency is difficult. It was but natural, therefore, that other types of equipment, progressing more rapidly towards economical and satisfactory performance, eclipsed this type of burner within a few years.

A modified form of heat-vaporizing burner, employing evaporative application in a different manner, has become a firmly established domestic burner. Figure 31 illustrates its general form. Here, a rotating distributor centrally placed spreads the oil over a refractory or alloy-steel hearth that is dish-shaped, so that the fuel comes in contact with the hot surface and is vaporized around the circumference of the firebox, resulting in a flame that travels upward around the walls of the furnace or boiler firebox.

This type, as shown in Fig. 32, is known as the vertical rotary wall-flame burner. Access of oil to the distributing mechanism is effected usually in one of two ways: upward through a hollow motor shaft from the burner base to a rotating head; or up through a stationary tube into a distributor cup which is rotated by the motor shaft. Whichever is used, from the burner head the oil is ejected horizontally and radially out toward the vaporizing ring. In a properly functioning plant, there should be no heat radiated from the walls to either the oil flowing out to the walls or the mechanism in the center. This is assisted by the fact that the characteristic flame of a wall-wiping burner is blue, so

that little heat is radiated from the flame itself. The temperature attained by the vaporizing ring can be controlled to some extent by the height to which it is built above the hearth for in that manner the cooling effect of the boiler walls is regulated. (This point must be borne in mind when installing in hot-air furnaces,

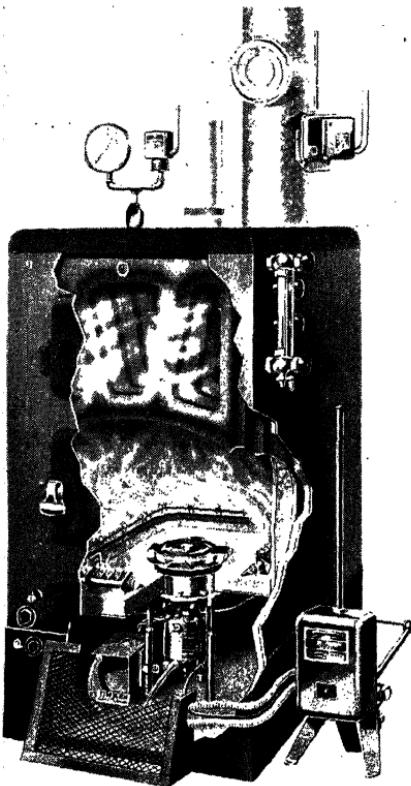


FIG. 32.—Installation of wall-wiping flame vaporizing burner. (*Courtesy of Timken-Detroit Axel Company.*)

for then the walls become much hotter than will be the case in steam or hot-water boilers.)

Ignition is accomplished in the wall flame burner by either gas pilot or an electric spark. In the former case, the gas is kept burning as a small pilot when the burner is off. Just before the burner starts up, this flame is expanded so as to heat up a portion of the refractory ring, in order that vaporization may commence as soon as the oil flow commences. When electric ignition is

employed, a small portion of the refractory ring is replaced by a carbex insert which serves as one electrode. The other electrode comes up through the hearth bottom and terminates near the carbex insert, the distance being called the spark gap. A short, hot spark is then thrown across the gap in order to secure the heat for the initial vaporization.

The air stream is supplied by the fan or impeller wheel which is also driven by the motor. This stream of air mixes with the oil as it travels outward to the fire ring, so that the vaporized oil is already in contact with air when gasification occurs. Hence this air might be considered as largely primary; a fact which accounts for the typical blue flame characteristic of wall-flame burners.

All vaporizing burners have one thing in common—the motor employed to drive the fan cannot readily be, and usually is not, used to drive an oil pump for delivering oil to the burner head. These burners therefore are usually arranged to receive oil from a supply tank from which oil will flow by gravity to the burner, or from a wall-mounted oil pump which is integral with a small supply tank from which oil will flow down to the burner. This latter type of pump is provided with a float switch to control the pump motor, and the level of oil is thus maintained within a few inches of a constant. When large tanks are connected to feed oil by gravity, provision must be made to feed at a fairly uniform rate—this can be had by the use of a constant-level device between the tank and the burner.

The wall-flame vaporizing burner has largely overcome many of the early faults of vaporizing types. It can be made to yield very satisfactory efficiency, and to operate without serious mechanical maintenance. It is however, restricted to burning either No. 1 or the better grades of No. 2 oil. The mechanism, being completely inside the boiler or furnace, is not readily accessible for adjustment or repair. In cost, this burner averages about the same as the vertical atomizing burner and somewhat more than the pressure atomizing burner.

As to capacity range, the type shown in Fig. 29 is limited to about 2 g.p.h. (300 sq. ft. steam, 960 sq. ft. hot water).¹ The

¹ Throughout this chapter, when burner capacities and similar matters are discussed, figures will be given in terms of gallons per hour actual burner capacity, but loads will be given in terms of square feet of steam or hot water

OIL BURNERS

pot type, Fig. 30, should logically be limited to 5 g.p.h. (1500 sq. ft. steam, 2400 sq. ft. hot water), although certain makes have been produced up to twice that size. Maximum capacity on the wall-flame type would logically be about 3 g.p.h. (900 sq. ft. steam, 1440 sq. ft. hot water).

Of recent years there has been a very marked progress made in the design and production of natural-draft burners for use with light fuel oil, but such burners are intended for special purposes rather than for central heating. This type of burner is considered in detail in Chap. XIX.

ATOMIZATION BY LOW OIL AND AIR PRESSURE

This very successful type of domestic oil burner was for a long time known and described as an emulsifying burner. Briefly, the original application of principle consisted in agitating together measured portions of air and oil so as to form what was thought to be a true emulsion, and then forcing this emulsion out through a nozzle as a finely subdivided spray.¹ However, closer observation and investigation showed that the air-and-oil mixture, while passing from the agitating chamber to the nozzle, tended to separate into two distinct streams—one being the air saturated with oil vapor and the other being oil containing all the dissolved oxygen it could hold. Finally, in passing through the nozzle, the two streams were again merged and subjected to

radiation (240 and 150 B.t.u. per hr., respectively) which the same number of gallons will carry after allowance has been made for combustion losses, pipe losses and pickup from cold start. Since grades of oil vary in heat content per gallon, and since burners and boilers vary in efficiency, it is necessary to arbitrarily establish a basis for expressing load capacity. This book will use 300 sq. ft. of steam radiation or 480 sq. ft. of hot water radiation carried per gallon of oil burned per hour, regardless of grade of oil, kind of burner or type of boiler or furnace. The reference to square feet of radiation signifies actual standing radiation connected to the heating system.

¹ A true emulsion consists of a mixture of two liquids, ordinarily immiscible (not mixing) with each other, but in which one liquid has been finely subdivided and its particles held in suspension by the other; or of a liquid and a solid insoluble in it, but in which liquid the solid is held in suspension in very fine particles; or of a liquid and an insoluble gas similarly held in suspension. Examples are: mayonnaise (two immiscible liquids), milk (water and butter fat, casein, etc.), and whipped cream (liquid and air).

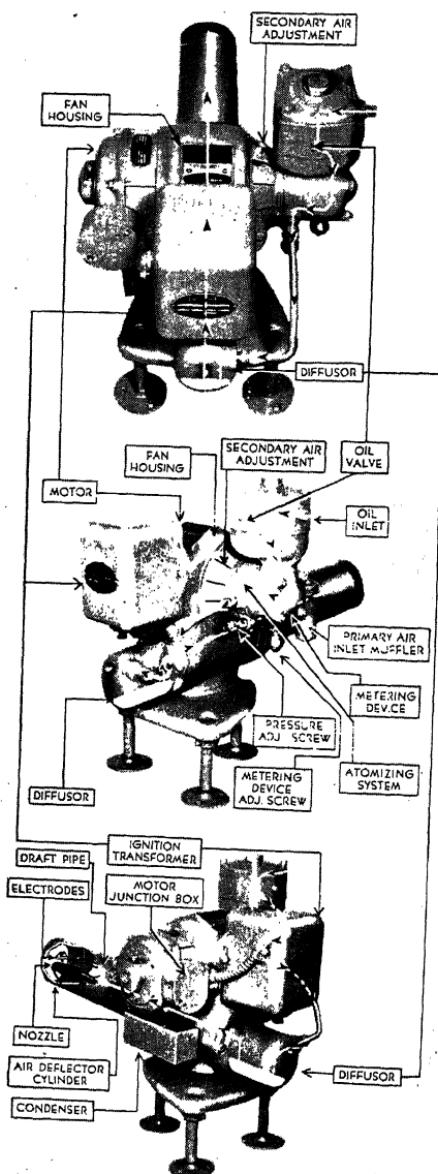


FIG. 33.—Low-pressure air- and oil-atomizing burner. (Courtesy of Williams Oil-O-Matic Heating Corporation.)

further agitation, to give the ultimate spray which issued from the nozzle.

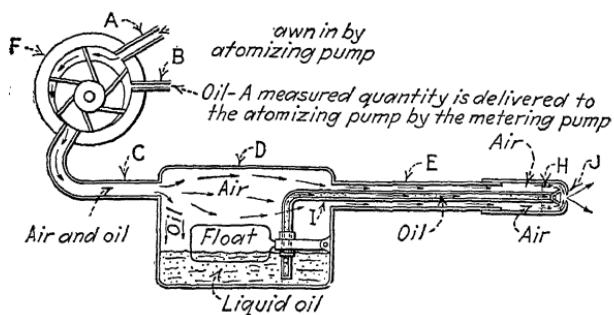


FIG. 34.—Diagram of working principle of low-pressure air- and oil-atomizing burner. A, primary air intake; B, oil intake; C, feed pipe; D, diffuser; E, air pipe; F, atomizing cylinder; H, oil spiral; I, oil tube; J, nozzle. (Williams Oil-O-Matic Heating Corporation.)

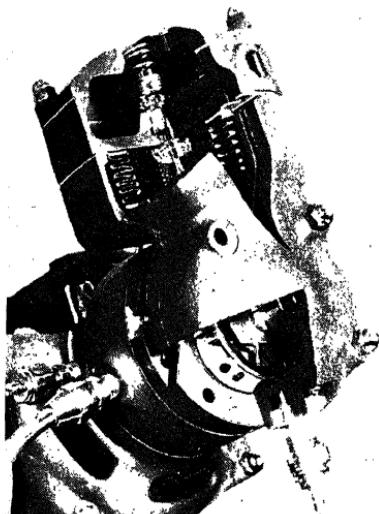
Once the true nature of what was occurring to the air and oil was understood and recognized, the equipment was modified to

conduct the air and oil separately from the agitating chamber up to the nozzle and there recombining the two. Figure 33 shows a typical burner of this classification, and Fig. 34 indicates in diagram the working principles. Oil is drawn up from the storage tank by the pump and discharged into a strainer, after which a division occurs and the surplus above that needed for the fire is returned to storage. The remainder, which is a carefully metered quantity because it is

rawn in by a metering pump, s delivered by the metering pump into the atomizing chamber where it undergoes agita-

FIG. 35.—Section through oil shutoff valve and metering pump. (Courtesy of Williams Oil-O-Matic Heating Corporation.)

tion with air. This mixture of oil and air is delivered under pressure to a receptacle where the two separate, except that the air now carries oil vapor and the oil has some air dissolved in



it. Each is conducted up to the nozzle, where they are again brought together and forced through the nozzle to effect final atomization. At no time during passage of oil through the machine does the pressure exceed 10 lb.

The amount of air introduced into the fuel for atomizing purposes is only a fraction of that required for complete combustion. The rest is blown in by a fan driven by the same motor which drives

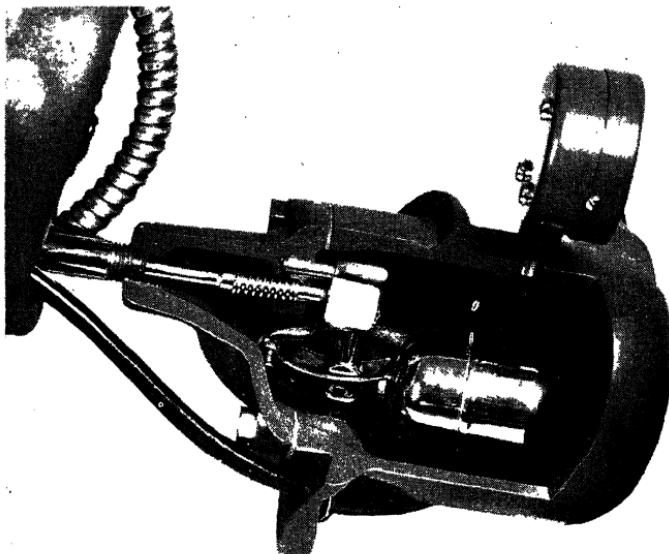


FIG. 36.—Section through diffuser. (*Courtesy of Williams Oil-O-Matic Heating Corporation.*)

the other moving mechanism. Thus in this type of burner it is seen that there are really two air supplies, primary and secondary.

Ignition on this type burner is usually accomplished by an electric spark, created by the high-tension electrodes near the burner tip. Regulation of fire size is had by adjustment of the metering pump, which is of a positive-displacement type. These burners regularly use No. 3 fuel oil, and in general can handle a fuel up to a viscosity of 70 Saybolt universal at 100°F. As a rule most of the applications of these burners do not operate at capacities greater than 15 g.p.h. (4500 sq. ft. steam, 7200 sq. ft. hot water). While, of course, the domestic sizes are only about one-fifth of that capacity, on the other hand there are sizes made up to 25 g.p.h. (7500 sq. ft. steam, 12,000 sq. ft. hot water).

When the fuel supply for this type burner is contained in a tank at a suitable elevation, the suction pump may be omitted, for oil will flow to the strainer and metering pump by gravity. In such case, a constant-level device is not needed, owing to the metering pump.

ATOMIZATION BY HIGH PRESSURE AND CENTRIFUGAL ACTION

The pressure-type burner is almost universally referred to as a gun-type oil burner. The design of this type of burner varies considerably with each manufacturer; however, the operating principle involved is essentially the same. This type of burner consists briefly of the following equipment: a base or frame (this may also include the draft tube of the burner) on which are mounted a motor and fan, a fuel pump, a pressure-regulating valve, a transformer, a primary strainer, means for ignition of the oil and a nozzle. These essential parts may be assembled in various ways to give different arrangements of the parts.

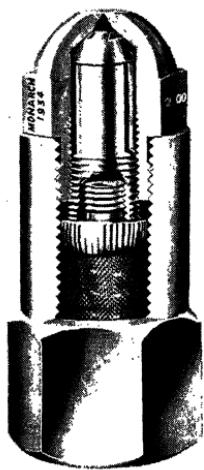


FIG. 37.—Section through pressure-atomizing oil nozzle.
(Courtesy of Monarch Manufacturing Works.)

Oil-atomizing nozzles are of two general types, of which the first is the conventional: (1) The nozzle which depends upon high oil velocities in the tangential slots in the nozzle thus causing a very rapid rotation of the oil before it passes through the orifice. In this type of nozzle the oil forms in a bubble at the end of the nozzle at low pressures and breaks out into a shiny cone-shaped film of vapor when the proper pressure is exerted on the oil. This type of nozzle is of the centrifugal-action type and is illustrated in Fig. 37. (2) The nozzle which combines centrifugal action with pressure at the orifice. In this type of nozzle the oil bubble referred to in nozzle Type 1 does not appear.

Air for combustion is furnished by a fan connected directly to the motor. A draft over the fire is necessary for all burners. Two-hundredths-inch (0.02-in.) draft is all that is necessary with this type of burner to develop its rated capacity.

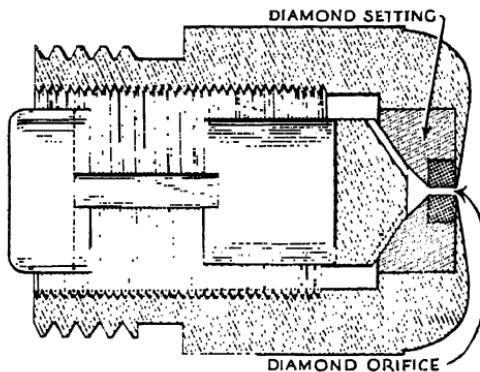


FIG. 38.—Section through pressure-atomizing oil nozzle with diamond tip.
(Courtesy of Balloffet Diamond Wire Dies Company.)

Ignition is almost universally of the electric type incorporating the use of a high voltage transformer and a spark gap, the spark gap to be just out of the path of the oil as it leaves the nozzle. Variations from this are sometimes used employing gas for igni-

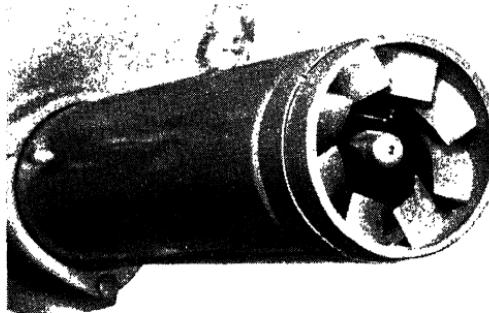


FIG. 39.—Air diffuser in opening of air tube.

tion, also a combination of the electric and the gas in which the electric spark ignites the gas.

Installation of the gun-type burner is usually quite simple if the manufacturer's instructions are faithfully carried out. Service on this type of burner is small and can be kept at a minimum if clean oil is used.

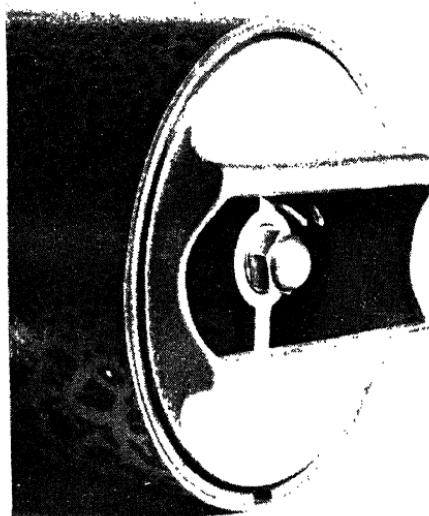


FIG. 40.—Air-tube mouthpiece. (*Courtesy of Harvey-Whipple Incorporated.*)

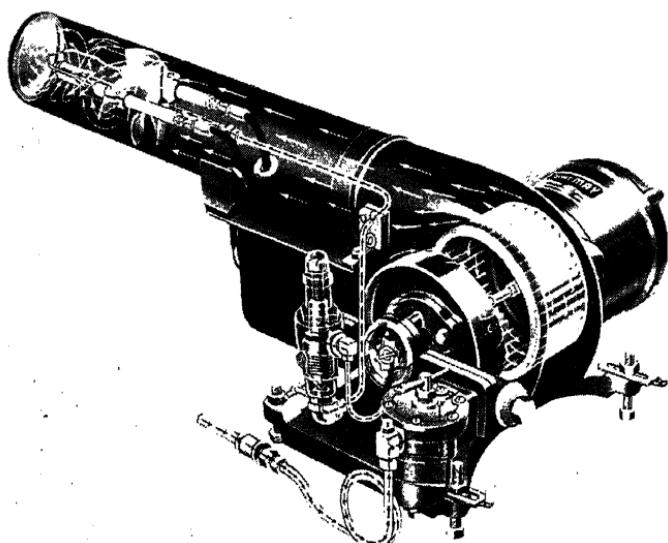


FIG. 41.—Phantom view of pressure-atomizing burner, showing paths of flow of oil and air. (*Courtesy of May Oil Burner Corporation.*)

Pressure- or gun-type oil burners can be made to operate at very high efficiencies if properly installed and adjusted. Since

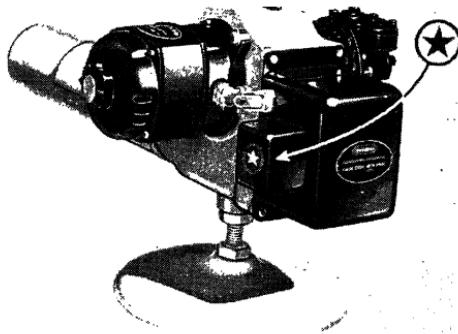


FIG. 42.—Typical pressure-atomozing burner, having special ignition-failure preventive device built into ignition transformer. (*Courtesy of Harvey-Whipple Incorporated.*)

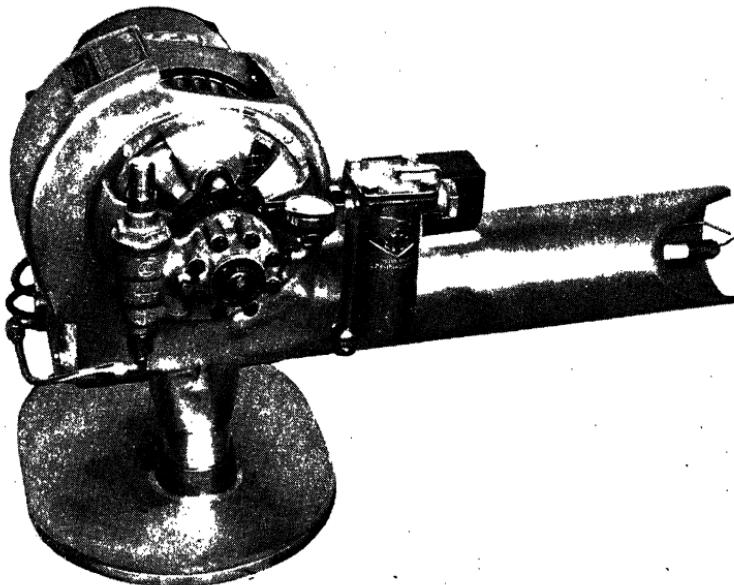


FIG. 43.—Typical pressure-atomizing burner assembly. (*Courtesy of Ace Engineering Company.*)

they are not so dependent on a strong draft as some other types, they are more stable in maintaining their settings. The limiting

factor in the capacity of this type of burner is the comparative cost of fuels. This has been the most successful type burner up to 10 g.p.h. On larger jobs the price of competitive fuels usually dictates the use of heavy-oil burners.

The need for quiet burner operation in residences requires low air velocities through the air tube. But good combustion necessitates intimate and thorough mixing of oil and air. To assist in accomplishing this at low air velocities, diffusers of various construction are used. Diffusers are arrangements of metal placed in the air tube, so shaped as to impart a whirling or rotating motion to the air stream. Sometimes two or more diffusers are used in sets. The resultant rotation of the air stream is supposed to induce more intimate mixing of air and oil (Fig. 39).

Gun-type burners are suitable only for use with distilled oils having a viscosity of not over 50 sec. at 100°F. Saybolt universal, in capacities up to 5 g.p.h. (1500 sq. ft. steam, 2400 sq. ft. hot water), and of not over 70 sec. at greater capacities, up to 10 gal. (3000 sq. ft. steam, 4800 sq. ft. hot water).

ATOMIZATION BY VERTICAL-ROTATING CUP

Essentially, a burner that atomizes oil by means of a vertically rotating cup comprises an electric motor mounted vertically, on top of which are the cup and a fan, and a means of bringing the oil up to the cup. The cup must have a lip so turned that as the oil is whirled off, the spray travels outward radially in an horizontal plane. The fan delivers the air stream just under and parallel to the oil spray. A refractory hearth must be constructed also in an horizontal plane to provide a suitable environment so that the oil can burn as a flat, thin flame.

Actually, the cup is responsible for most of the atomization, the air stream serving more as a means of conveying the spray after atomization has been accomplished. The rapid rotation of the cup causes the fuel to spread out as a thin film on the cup surface, and as the oil particles in turn reach the cup edge, the final motion of the cup, as the fluid ceases contact, is one of atomization, *i.e.*, the physical state is changed from that of a continuous fluid film to that of a fine spray of minute discrete liquid particles.

In this type of burner the fan does not have any considerable capacity, in fact, the air supply is largely induced from natural

draft. Hence the burner capacity is determined almost entirely by the cup. The atomizing capacity increases directly with the diameter of the cup, and directly with the speed, but inversely with the viscosity of the fuel.

The vertical atomizing cup burner is capable of attaining high operating efficiency. Further, this operating efficiency is not

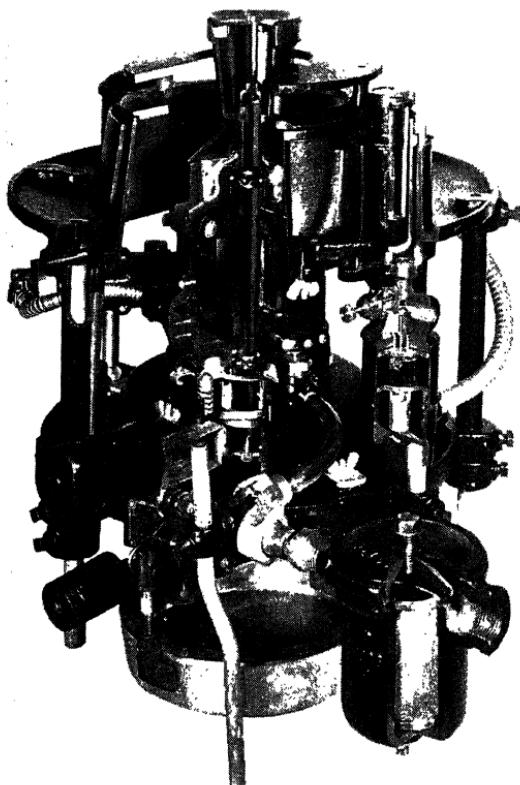


FIG. 44a.—Vertical rotary-cup atomizing burner, 10,000 r.p.m. (*Courtesy of Automatic Burner Corporation.*)

subject to wide disturbance by improper installation, at least not to the extent that is perhaps possible with other types. If this machine is correctly manufactured, as it usually is, as an integral mechanism that can be readily installed by men with only limited knowledge or training in heating or combustion, it will usually deliver a satisfactory efficiency.

Ignition on vertical-cup atomizing burners may be expanding gas, electric-gas or electric. An expanding-gas pilot is an arrangement which permits a small gas flame to burn continuously. When the burner is about to start, a solenoid gas valve opens and permits a full flow of gas, thereby greatly increasing the size of the gas flame. As soon as the ignition of the oil is accomplished, the solenoid closes and the gas flow drops

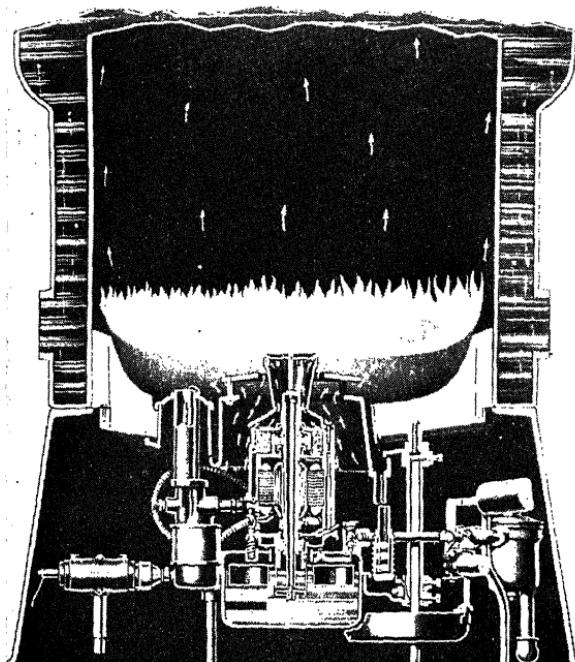


FIG. 44b.—Setting of vertical rotary-cup atomizing burner in round heating boiler. (*Courtesy of Automatic Burner Corporation.*)

to the small pilot again. In electric-gas ignition, the gas together with an electric spark comes on when the burner starts, the spark ignites the gas, the gas ignites the oil and, as soon as ignition is complete, both spark and gas flow cease. In electric ignition, the spark alone is used to ignite the oil. Usually, this spark comes on with the burner starting and ceases after the oil is burning. Sometimes, however, the spark continues as long as the burner runs.

Most burners of this type will burn No. 3 oil, but experience shows that unless the oil is exceptionally clean and of only

moderate viscosity (about 60 Saybolt universal at 100°F.), it is safer to use No. 2. The entire burner mechanism is located inside the boiler, so that complete removal is usually necessary for either cleaning or adjustment. This may not be a tedious or laborious process, but even so it is desirable to use as clean

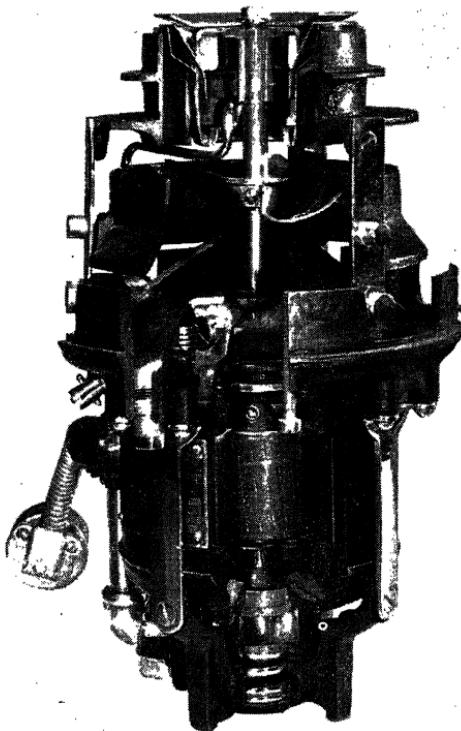


FIG. 45.—Vertical rotary-cup atomizing burner, 3500 r.p.m. (*Courtesy of Automatic Burner Corporation.*)

an oil as possible and so avoid the necessity for removing the burner for cleaning.

Successful application of this type of burner may be had in sizes up to about 10 g.p.h. (3000 sq. ft. steam, 4800 sq. ft. hot water).

Vertical-cup atomizing burners are usually designed for gravity feeding of the oil, as in the case of vertical-rotating vaporizing burners. (For discussion on methods of feeding oil, see page 109.)

ATOMIZATION BY HORIZONTAL-ROTATING CUP

The actual operating principle of the horizontal-rotary burner is considered in detail in Chap. VII. It need only be mentioned here that in smaller sizes the horizontal rotary has had a limited application to domestic heating and to small industrial applications, where for one reason or other its use has been indicated as best.

The horizontal rotary is especially useful in territories where available fuels are too heavy for other types of small burners, or for applications where an inexpensive installation is desired in such diverse cases as bakers' ovens, tailor's pressing-machine boilers, boiler-burner units, etc.

Ignition on a domestic horizontal rotary is most satisfactorily accomplished with expanding-gas or electric-gas systems. Straight electric ignition is also used, but not so successfully as the former two. On electric ignition, with high flash-point oils, it is usually desirable to use a 15,000-volt transformer to create the ignition spark.

This kind of oil burner is somewhat more noisy in operation than pressure-atomizing and vaporizing burners. Usually, gravity oil feeding is used. Special provision must be made to support the burner in place by means of a steel (or other suitable material) plate, on which the burner usually is hinged so that it may be swung open to allow inspection and cleaning of the air nozzle and oil atomizer. A firebox of firebrick or plastic refractory must be constructed inside the boiler or furnace. Largest sizes are about 4 g.p.h. (1200 sq. ft. steam, 1920 sq. ft. hot water), and the customary fuel used is No. 3, although some grades of No. 4 can be handled, especially if preheated.

ATOMIZATION BY AIR STREAM

The true low-pressure atomizing nozzle is strictly a commercial or industrial burner, even though some use of it has been made in domestic practice. Full consideration of it is given in detail in Chap. VIII. Another form or variation of form of the air nozzle is the so-called vacuum-pressure nozzle, which perhaps can be more adequately described by the term aspirating nozzle. This is somewhat more adaptable to domestic use, but even so, since by far the greater number of its applications and installa-

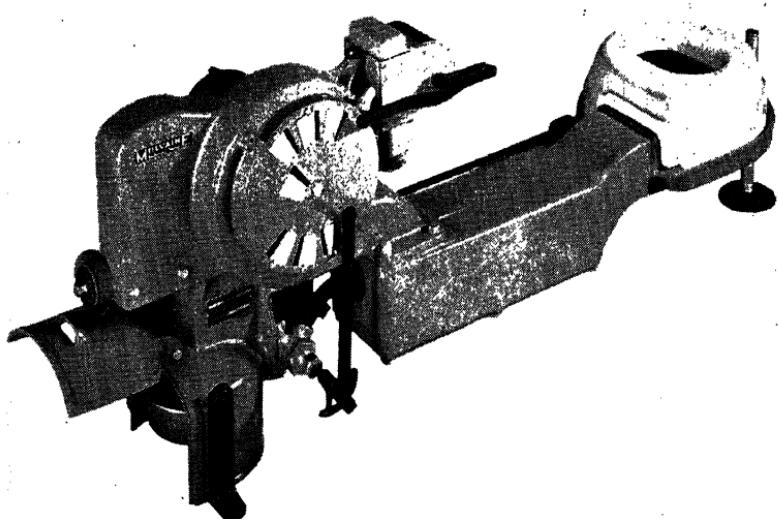


FIG. 46.—Vaporizing burner with high-low flame adjustment. (*Courtesy of McIlvain Burner Corporation.*)

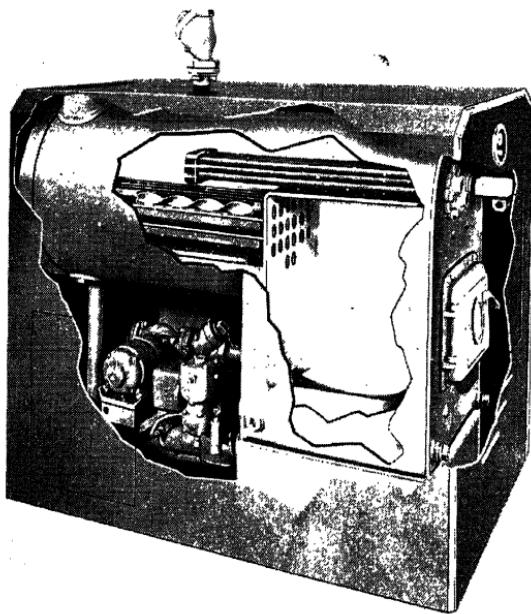


FIG. 47.—Steel-boiler-burner unit. (*Courtesy of Orr and Sembower.*)

tions lie in the industrial-process field, detailed consideration of it is likewise reserved for Chap. VIII.

SPECIAL TYPES

Certain kinds of oil burners embody two or more of the fundamental methods of atomization. One form of burner employs a

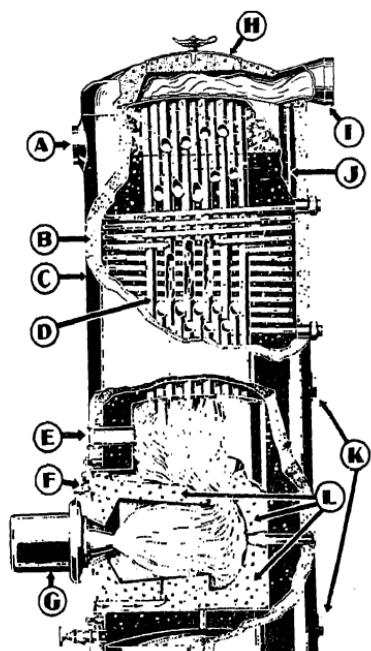


FIG. 48.—Vertical steel-boiler burner unit. *A*, pressure gauge; *B*, mineral wool insulation; *C*, steel jacket; *D*, fire tubes; *E*, peep hole with pyrex door, *F*, secondary air inlet and adjustment; *G*, horizontal-rotary burner; *H*, removable flue cover; *I*, chimney connection; *J*, boiler shell; *K*, clean-out plugs; *L*, precast refractory lining. (*Courtesy of S. T. Johnson Company.*)

successfully burn fuel oils heavier than No. 4. It is a rather complicated mechanism and the maintenance expense is high. For domestic use, it can readily be used in capacities ranging up to 10 g.p.h. (3000 sq. ft. steam, 4800 sq. ft. hot water).

vertical atomizing cup which is driven from a horizontal motor by means of a suitable gear box. A high-capacity fan is driven from the vertical cup shaft. A refractory hearth with a high side wall is provided. Atomization is initiated by centrifugal action at the cup perimeter, which throws off the oil radially in the form of a coarse spray. The strong air stream immediately carries the spray to the outer ring of the firebox, at the same time assisting in breaking down the oil particles. At the ring, vaporization by heat completes the process of preparing the oil for combustion. The flame has the general shape of the straight-vaporizing wall type above described. Ignition is preferably expanding gas, although electric-gas is available

grade of oil best suited is No. 2. This type of burner has been made in commercial sizes, but as such is generally unable to

heavier than No. 4. It is a rather complicated mechanism and the maintenance expense is high. For domestic use, it can readily be used in capacities ranging up to 10 g.p.h. (3000 sq. ft. steam, 4800 sq. ft. hot water).

A modification of the original pot-type vaporizing burner described on page 106 embodies certain refinements and is made today as a continuously operating high-low domestic burner. Its general appearance is indicated by Fig. 46. In order to place this burner in operation at the beginning of the heating season,

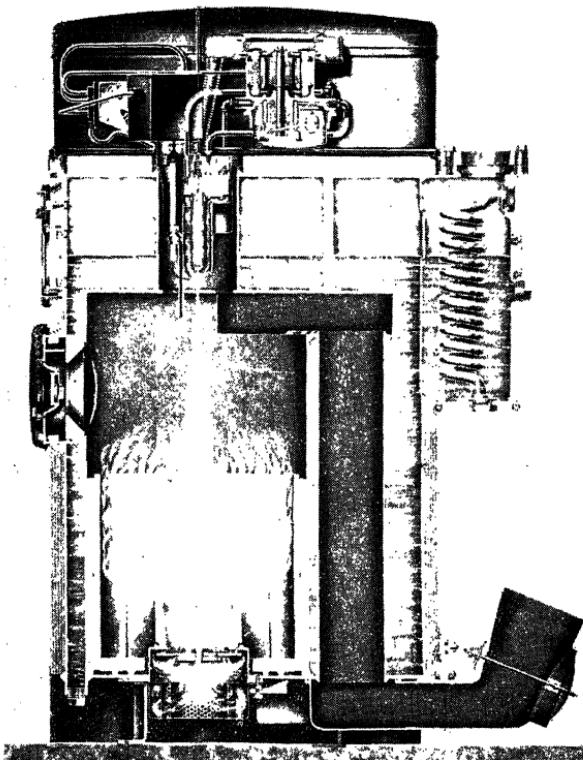


Fig. 49.—Steel-boiler vertical-burner unit. (*Courtesy of General Electric Company.*)

heat must be applied to the combustion head, possibly from an outside source such as a gas flame. Thereafter the oil flame is continuous, being varied in size by the regulator mechanism which has four positions, each position giving a fixed rate of oil flow and air. The feed is gravity, there being a constant-level device built into the mechanism. A special thermostat actuates a motor which determines the position of the four-speed control.

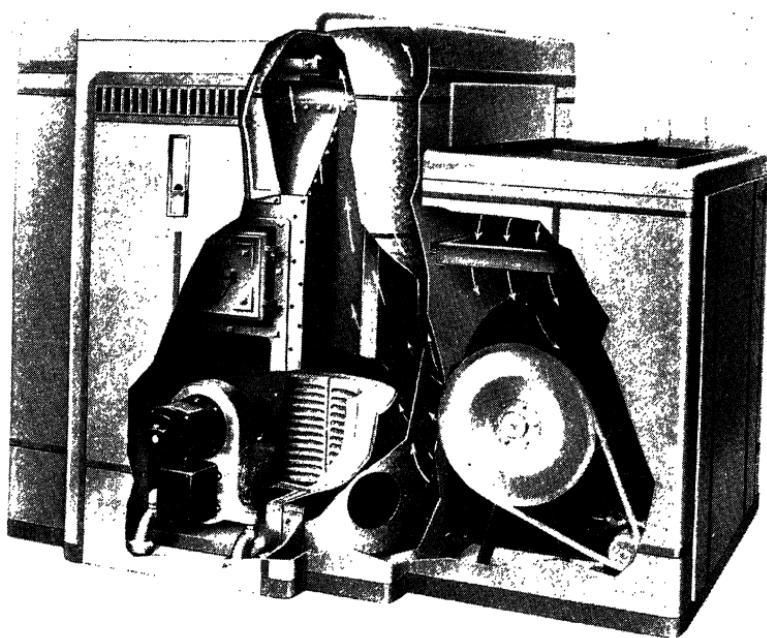


FIG. 50.—Warm-air furnace-burner unit. (*Courtesy of Fox Furnace Company*)

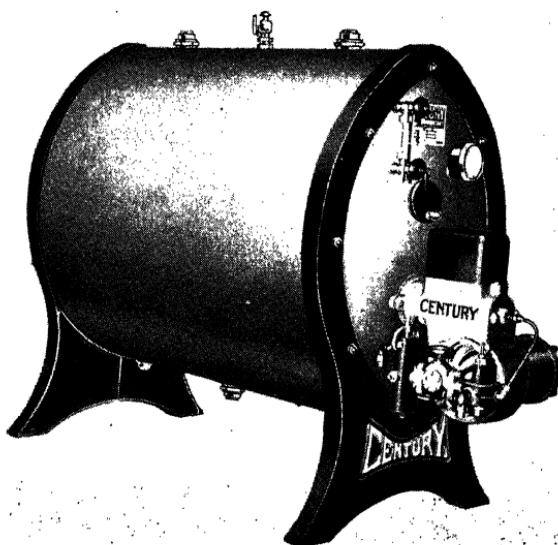


FIG. 51.—Scotch-type boiler-burner unit. (*Courtesy of Century Engineering Company*)

BOILER-BURNER AND FURNACE-BURNER UNITS

During the past few years a pronounced tendency has arisen, in cooperative efforts between boilermakers and burner manufacturers, to offer a product combining boiler or furnace and oil burner into a single unit, to be sold and installed through established channels in the merchandising and contracting field. These units are designed not only from the viewpoint of maximum utility, but also with a view to incorporating harmony of design and pleasing appearance.

There are numerous advantages to such combination units over the conventional application of oil burners to heating devices primarily designed for coal burning. Thus the combustion space may be so proportioned as to yield the most satisfactory results from an oil fire rather than from a coal fire. Provision can be made for extending the boiler or furnace jacket so as to include the burner mechanism, thereby yielding a fully enclosed heating unit. In boilers, a submerged hot-water heating coil is generally furnished. With warm-air furnaces, it is general practice to have the unit embrace circulating fan, air washer, humidifying equipment, etc.

Typical boiler-burner and furnace-burner units are illustrated in these pages. While they vary considerably in boiler type, it is worthy of note that most of them possess symmetry of style.

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CHAPTER VII

COMMERCIAL OIL BURNERS

By the classification commercial, this book intends to imply that a burner has general application in the heating field for the heating of most buildings other than residences or small apartments or other small buildings. It is, of course, difficult, in fact almost impossible, to draw a fine line of distinction between what should be called a large domestic installation and what should be termed a small commercial one. There is rather a broad overlapping of application. But if an average were sought, it would probably be found that most domestic burners cease to find use in plants having more than 2000 sq. ft. of steam radiation, or equivalent, and that few commercial burners are used in places running under 1500 sq. ft. of steam or equivalent.

Viewing the matter from the grade of oil handled, a much clearer distinction may be drawn. The great majority of domestic installations operate with No. 3 fuel oil or lighter; commercial installations burn usually either No. 5 or 6. Only in the case of No. 4 oil is there a question of burner type, and this particular grade of oil is being used in steadily diminishing quantities in successive years—a mute but powerful testimony to the fact that No. 3 oil represents the low limit of domestic fuels and No. 5 the upper for commercial.

Commercial oil burners are classified as follows:

1. Horizontal rotary.
2. Air atomizing.
 - a. Low pressure.
 - b. High pressure.
3. Steam atomizing.

HORIZONTAL-ROTARY OIL BURNERS

Within recent years, this type of burner has easily become the leading and most successful of all commercial-type oil burners. The reasons for its success are manifest. Mechanically it need not be complex. It can be made as a compact integral unit,

self-contained and lending itself to simple installation methods. It is adaptable to single-unit installation or multiple-unit installation. It is equally useful in manual, semiautomatic or full-automatic control. It burns heavy oils at satisfactory efficiencies. It has a wide range of capacity in any given size. Its flame can be altered from a short, bushy, fanlike fire to a long, straight-shot fire by comparatively simple adjustment or change of nozzle. Its operating expense for power is low, and maintenance costs are also low.

There are, of course, many variations of form and design of the horizontal-rotary burner, resulting from design and construc-

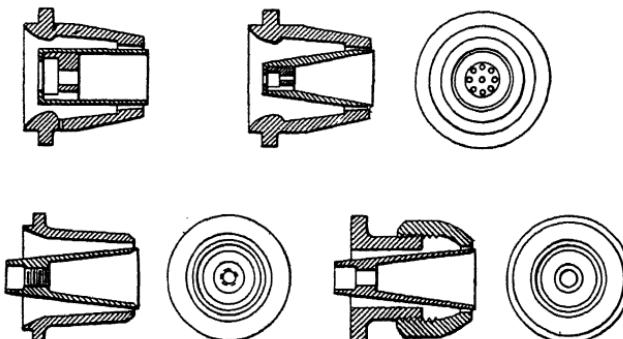


FIG. 52.—Atomizing cups and air nozzles for horizontal-rotary burners.

tion ideas and opinions of different manufacturers, but in spite of this, there is surprising uniformity of flame application throughout. The standard integral unit consists always of the same basic parts—electric motor with extension shaft, on which are mounted fan, atomizer and pump-drive gears. Very often, especially when it is desired to employ two or more burners in the same boiler room, the oil pump is not integral with the burner, but oil is supplied to the burners from a separately driven oil pump. Less frequently, the fan is also lacking from the motor shaft, and air is delivered around the atomizer from an external source. Still less often, the atomizer, instead of being motor driven, is propelled by an air turbine. It follows, therefore, that the single characteristic feature of the horizontal-rotary burner is the oil-atomizing cup and the enclosing air nozzle, from which the oil spray is delivered into the furnace or combustion space to yield the horizontal oil flame.

Since the atomizing cup is the heart of this burner, it is worth while considering it in some detail. Figure 52 shows an assortment of atomizing cups in section, with the method of introducing the oil to the cup indicated in each case. It should be noted that the method of introducing the oil to the cup will affect the cup design. When oil comes in through the center, by way of the shaft, the cup sides terminate at the base, but where oil is fed in from a pipe, the cup sides must be extended beyond the base in order to form a receptacle into which the oil can be delivered. In this latter case, the base must be perforated in some manner to permit the passage of oil up into the cup. When heavy and viscous oils are being handled, this feature may be objectionable, for the openings through the base may become clogged with sediment or even with congealed oil if allowed to cool there after the burner has been stopped.

Roughly, the relationship existing between the capacity of a cup and its size is that the perimeter equals approximately the square root of the number of gallons per hour that can be atomized. This rule is subject to modification, however, by other factors, such as the pressure developed in the air stream which must be used to carry away the oil spray and, of course, the cup speed. The cup length must be sufficient to permit the oil to spread out as a film uniformly over the cup surface before reaching the atomizing edge. The effectiveness of atomization depends largely upon the oil viscosity, and also upon any previous atomizing effect to which the oil has been subjected before reaching the cup.

The standard speed of atomizing cups, in motor-driven burners, is 3450 r.p.m.. In air-turbine-driven burners, this speed may be greatly increased, sometimes attaining 7000 to 10,000 r.p.m. It is doubtful, however, if much is gained by using speeds above 3450, for whatever might be gained in capacity on any given cup could be secured by using a larger diameter cup at 3450 r.p.m., while on the other hand the problem of adequate bearings becomes much more aggravated at the higher speeds.

The shape of the cup is influenced mostly by the desired shape of fire, and the fact that the fire shape can be so controlled constitutes one of the major advantages of the horizontal-rotary burner. In general, a true cylindrical bore to the cup will yield the so-called straight-shot fire, which has the smallest amount of

taper, while a cone-shaped cup yields a fire with considerably more taper. For the same amount of oil being burned, the greater the taper the shorter the fire. The selection of cup shape therefore should be guided by the shape of the combustion chamber which will be available in any particular installations.

An indispensable accessory to the atomizing cup, and one that must always be present as complementary to it, is the air nozzle which surrounds the cup. The air nozzle serves these functions: it partially regulates the amount of air delivered as primary air; in common with the atomizer itself, it partially controls the shape of the fire; and it especially determines the closeness with which the fire can be held back against the atomizer. The natural tendency of the fire is to commence at the cup, for as soon as the oil leaves the rotating edge of the atomizer it is in combustible condition. But the velocity imparted to the oil spray by the cup and by the air stream causes the fire to commence at a distance of several inches from the cup itself. If the oil supply is cut too low, the air stream may carry the oil clear to the back-wall of the firebox before it burns. Under normal conditions, however, oil and primary air are increased or diminished in synchronism so that the fire may burn along the length of the firebox. Whenever it is desired to reduce a fire size so low that it begins to break away from the cup and the tendency cannot be overcome by closing the primary air damper, then the air-nozzle size should be reduced. Conversely, if the fire size must be increased beyond the normal capacity of a burner, with resultant smoke, the fire may be cleared up by increasing the air-nozzle diameter. Some manufacturers of rotary burners turn out burner sizes which differ from each other only by dimensions of the cup and air nozzle, *i.e.*, the motor, fan, pump and other parts being identical.

The air nozzle on a horizontal-rotary oil burner is always so made and attached to the burner that it may be readily removed for either cleaning or replacement if it is desired to alter or modify the fire shape or size beyond such alteration as can be secured by modification of the oil feed or primary air damper. Sometimes this feature of the burner, the air nozzle, is made adjustable, within narrow limits, by having the air-nozzle casting screwed onto the front fan housing by means of a fine-pitch thread. While undoubtedly this method does permit of some latitude in

securing primary air variation without the necessity of changing air nozzles, it has been found that the respective lengths of air nozzle and oil atomizer are related to each other very closely, and that very close limits must be maintained in the length of oil atomizer which protrudes out beyond the air nozzle. Hence, should the thread on the air nozzle permit this limit to be exceeded, a very poor fire would result.

In certain types of horizontal rotary, notably where vanes are inserted into the air nozzle to impart a counter-rotary motion to the air stream in opposition to the rotary motion of the oil spray, the lengths of air nozzle and oil atomizer are very closely the same, *i.e.*, the two terminate in the same vertical plane. This represents one extreme. The other extreme is about $\frac{1}{4}$ -in. additional length of the oil atomizer beyond the air nozzle. The atomizer should never terminate inside the nozzle, and rarely indeed should it be longer than $\frac{1}{4}$ in. beyond the nozzle.

The motion imparted to the oil spray by the cup tends to move it in a direction perpendicular to the cup axis. The air stream tends to carry the oil spray forward, and also in a slightly rotating path as a result of impetus from the fan. The actual path taken by the oil spray, and hence the shape of the fire, is a resultant of these separate imparted motions. It follows that fire shape can be controlled to some extent by the air-nozzle design with any given oil-atomizer size and shape. But in all cases it is important that the width of the air stream at the tip shall be sufficient to prevent oil particles from passing entirely through the air stream into the quiescent air beyond, for then the oil would fall to the firebox floor and smolder. A good base design for average conditions is to have the ratio of air-nozzle diameter to the oil-atomizer diameter at about 1.2. For increased firing rate it can be made greater; for lower firing rates the ratio can be less. For the same fan and atomizer, an increase in air-nozzle diameter tends to lower the air pressure at the tip; nevertheless, the velocity from that point onward into the firebox becomes greater, because a greater amount of air then flows through the firebox, which, of course, has a constant cross section. This increased firebox velocity of the air-oil mixture will overcome any ill effects of diminished air pressure at the tip resulting from the enlarged air-nozzle diameter. The exact converse occurs in

reducing the air-nozzle diameter for the purpose of decreasing oil-firing rate.

The third distinguishing feature of a horizontal-rotary oil burner is the method of introducing oil into the cup. There are two general methods, each in turn having variations. These general methods might be termed the side-tube and the hollow-shaft methods. In the first, the fuel is brought up to and into the cup by means of a tube or pipe from a point roughly perpendicular to the shaft and atomizer, usually along the front fan housing. This tube or pipe has a short tip which bends over and into the cup so that the fuel may be discharged into the cup

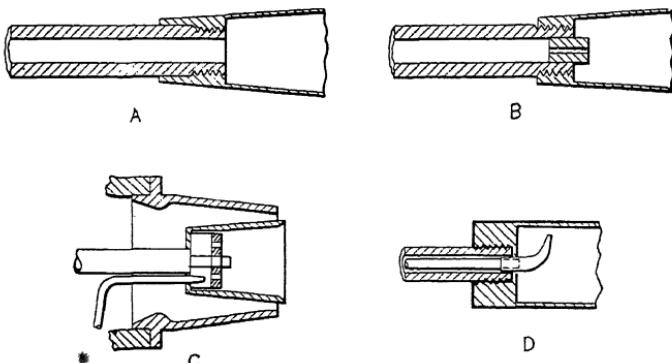


FIG. 53.—Methods of feeding oil to atomizer. *A*, oil feed to cup through hollow shaft, unrestricted opening; *B*, oil feed to cup through hollow shaft, drilled plug in opening; *C*, oil feed to cup through side tube; *D*, oil feed to cup through stationary tube in hollow shaft.

and be retained there by a ridge at its back end. The rotary motion of the cup then causes the oil to move forward, spreading as it does so as to form a film over the cup surface which can then be whirled off the cup edge and atomized. As mentioned before, in this method, the cup must be attached to the shaft by means of a base which must be kept forward a sufficient distance to allow for oil delivery into the cup. This base, necessarily perforated so that oil may pass through it, tends to restrict the formation of the film over the cup surface and also offers opportunity for gum and sediment to collect, thereby necessitating more or less frequent cleaning. In the second method, the hollow-shaft system, oil is conveyed directly through the entire length of the burner main shaft, from a point in back of the motor, up into the cup. The oil may pass through in direct

contact with the shaft itself, or it may go through a stationary tube which reaches straight through the shaft. In the former case it is necessary to provide a suitable packing gland around the back end of the shaft so that the fuel oil may enter it and be retained within the gland during passage. In the latter case, no gland is necessary. When the fuel is taken directly through the shaft, without the use of the stationary tube, it is usually found that the agitation thus imparted to the oil helps considerably in its atomization, provided that the front end of the shaft is left open to atmosphere. Sometimes, however, a plug with a small drilled opening is screwed into the shaft at this point to prevent drooling of oil when the burner stops running. This plug tends to nullify any advantage which might have been gained from the agitation imparted by the oil's whirling with air in the shaft bore. With either type of hollow-shaft feed, stationary tube or whirling oil, the cup shape is about the same and is somewhat simpler than the cup which must be used with the side-tube feed. Another advantage to the favor of the hollow shaft is that the cup screws onto the shaft with a relatively large connection, while the side-tube cup must use a much smaller connection to the shaft. The more rigid this connection is, and the greater the diameter of the shaft itself, the better likelihood there is that the cup will turn true with the shaft. On the whole, therefore, the hollow-shaft has advantages over the side-tube method of feeding oil into the cup. A variation of the hollow shaft is to have the shaft hollow only part way, and to introduce the oil into the shaft at a point along its length. This involves the use of a complicated oil-sealing arrangement.

Since the horizontal-rotary burner utilizes two forces to effect atomization—rotating cup and air stream—the technique of air-stream application must also be studied. The importance and use of the air nozzle have already been discussed; the design and construction of fans is an extensive subject, and rather than confuse the logical development of this type of burner at this point by introducing a lengthy treatment of the fan, this subject is treated in Chap. IX. It is more important here to observe the intensity of air stream required; the means of furnishing the air stream can be taken up later. Table XIV shows a chart for determining the required air pressure and volume and other pertinent data on primary air supply.

TABLE XIV.—DATA FOR DESIGN OF ATOMIZERS AND NOZZLES AND AIR PRESSURES ON HORIZONTAL-ROTARY OIL BURNERS

Burner capacity per hour, gal.	Oil atomizer diameter, in.	Air-nozzle diameter, in.	Ratio of diameters	Air-opening area, sq. in.	Required primary air at 200 cu. ft. per gal., cu. ft.	Velocity of air at nozzle, ft./min.	Pressure of air stream at nozzle, in. water
8	0.9	1.17	1.3	0.44	1,600	8,717	4.75
15	1.23	1.56	1.27	0.72	3,000	10,000	6.25
25	1.59	1.96	1.23	1.04	5,000	11,517	8.25
40	2.01	2.42	1.21	1.48	8,000	12,960	10.5
70	2.68	3.18	1.195	2.34	14,000	14,617	13.0
140	3.77	4.4	1.17	4.01	28,000	16,700	17.5

NOTES: Oil nozzle diameters are related to gallons per hour by the ratio

$$D = \frac{\sqrt{g.p.h.}}{\pi}$$

Static no delivery (S.N.D.) pressures developed by the fans will be about 1.3 as much as the above indicated air pressures at open-nozzle conditions.

For single-unit installations of this type burner, by far the simplest means of driving the oil pump to secure delivery of oil to the cup is from a set of gears, worm and drive, working from the main shaft of burner. Figure 54 illustrates the common method of attaching the gears, pump and other required apparatus incident to pumping oil. Pumps and related equipment, such as strainers, valves, pressure regulators, metering devices, etc., are discussed in Chaps. XII and XIII. It is almost universal practice by manufacturers to equip rotary burners with a swivel arrangement that permits of swinging the burner into and out of firing position to allow of quick and easy inspection of the atomizer and access to the entire burner mechanism without the necessity of disconnecting either oil or permanent electrical connections. The oil is therefore handled by swivel joints that have packing glands, or by flexible oil tubing, between the rigid oil piping and the pump and burner itself. The same is true of pumpless units, which are intended primarily for multiple installation where a single pumping source can deliver fuel to two or more burners.

The most common use of the horizontal-rotary burner is for central heating plants of buildings, and in this application the

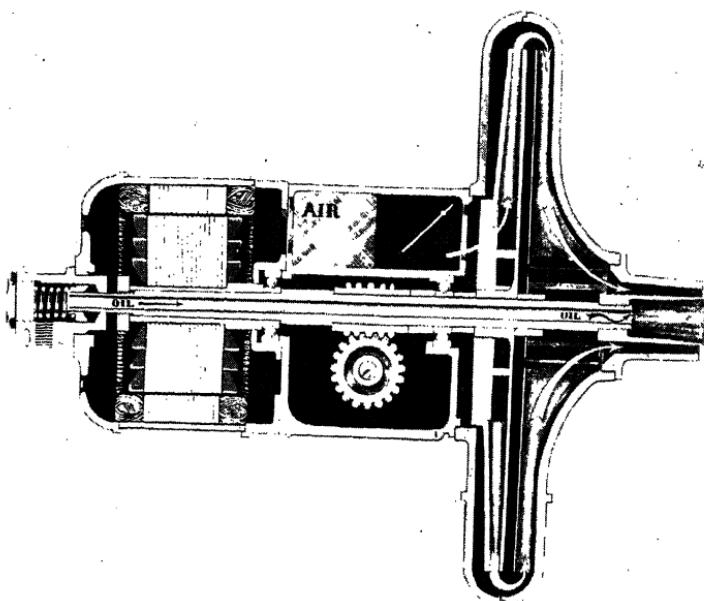


FIG. 54.—Conventional method of mounting motor, fan, atomizer, and pump-drive gear on single shaft. (*Courtesy of Ace Engineering Company.*)

burners are usually integral units, including pumps. Within

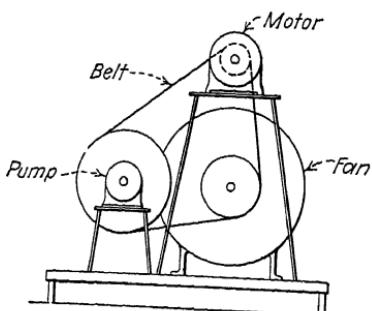


FIG. 55.—Optional method of belt drive from motor to oil pump, fan and atomizer on horizontal-rotary burner.

recent years the bulk of these installations have been made for automatic operation, with gas or electric-gas ignition, the primary control being either room-thermostat, boiler-limit or furnace-limit switch or some kind of mechanism for proportioning burner-operation periods to building heat requirements. In the average case, there is little need for complicated control mechanisms, and this intermittent type of operation yields very satisfactory results.

Not infrequently, however, these burners are installed for semiautomatic use, where ignition is accomplished by hand and

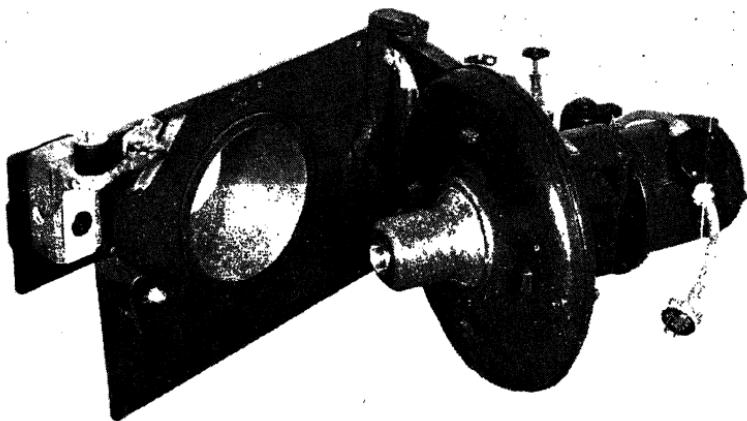


FIG. 56.—Horizontal-rotary oil burner with expanding-gas ignition. (*Courtesy of Ace Engineering Company.*)

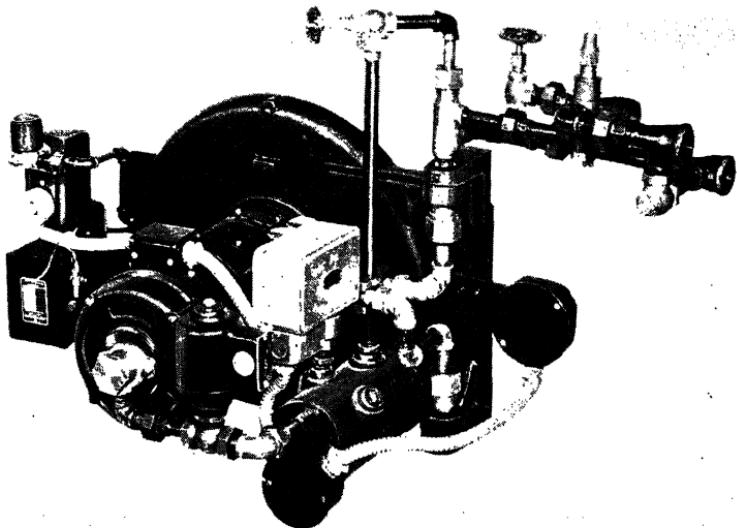


FIG. 57.—Horizontal-rotary oil burner with electric-gas ignition and electric oil heater. (*Courtesy of Ace Engineering Company.*)

the burner continues in operation until again stopped manually, except that in some cases operation may be interrupted by some

limit device such as a low-water safety or combustion safety cutout—but in case of such abnormal interruption the burner

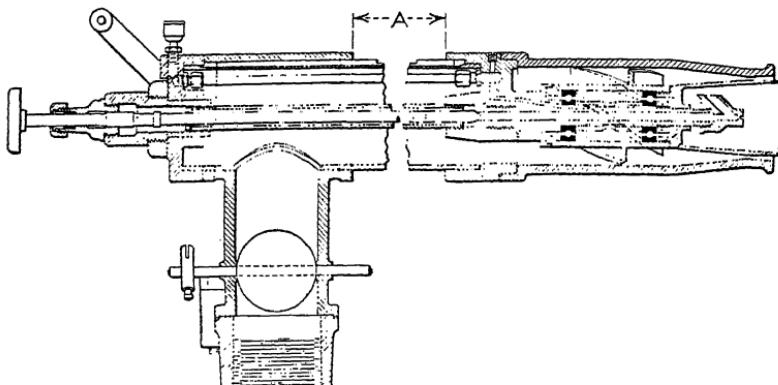


FIG. 58.—Horizontal-rotary oil burner employing air-turbine-drive for atomizing cup. (*Courtesy of Simplex Oil Heating Company.*)

remains off until again started by hand. It is during the period of normal operation that the semiautomatic feature arises, for

it is very seldom that a burner can be adjusted to a firing rate which will just handle the boiler or furnace load, or so that the heat input just balances the load output. It is necessary that the burner be equipped with a means of graduating the fire size in response to some feature that characterizes the load itself, either boiler steam pressure, or hot-water temperature, or furnace-bonnet temperature, in a manner approximating load fluctuations. The most important requirement of the burner adjusting mechanism is that the oil and air supply be graduated in synchronism with each other, so that excess air or inadequate air supply be avoided.

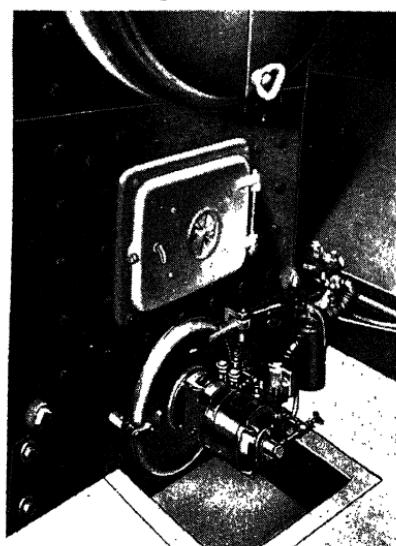


FIG. 59.—Typical installation of horizontal-rotary burner on steel heating boiler. (*Courtesy of Ace Engineering Company.*)

This system of burner regulation is often coupled up with an electrical control device.

The entire subject of electrical-operating controls, and safety controls, for automatic operation, is covered fully in Chap. X. Other control methods are discussed in Chap. XV.

A firebox or combustion chamber must be constructed within the boiler or furnace being fired. Because of the great variety of boiler and furnace types and forms in use, combustion-chamber design is a fairly involved subject. A complete discussion of the subject will be found in Chap. XIV.

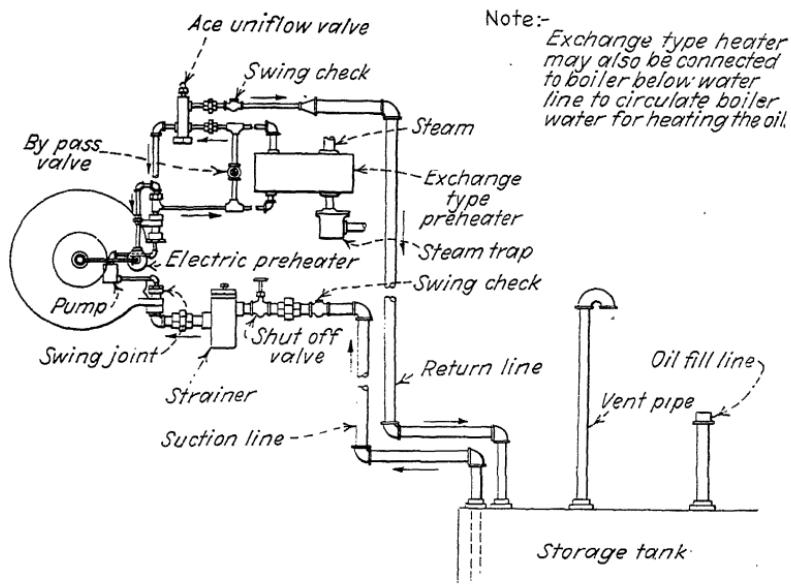


FIG. 60.—Typical oil-piping connections for horizontal-rotary burner with electric and indirect oil heaters.

Most horizontal-rotary burners intended for commercial work are designed to burn No. 5 oil without preheating and No. 6 oil preheated. However, some qualification of this statement is necessary. In the first place, the present limitations of viscosity of these two grades as set up by Commercial Standards 12-35 of the Bureau of Standards are broad; No. 5, under those specifications, may vary from 25 to 100 sec. Saybolt furol at 122°F., while No. 6 viscosity may be between the limits of 100 and 300 on the same scale. These values correspond roughly to 500 and 2000 for No. 5 and 2000 and 6000 for No. 6, seconds Saybolt universal at 100°F. As a matter of fact, even the larger sizes of horizontal-rotary burners will have difficulty in burning cold

No. 5 fuel oil when the Saybolt universal viscosity at 100°F. exceeds 600, and at, or above, 1000, it is about impossible even to conceive of such operation. Hence it is safe to say that all grades of No. 6 and any grade of No. 5 having a Saybolt universal viscosity above 1000 at 100°F. will require preheating for use in a horizontal-rotary oil burner.

When these burners are intended to burn No. 6 oil automatically, it is quite necessary that an electric heater be provided to furnish hot oil for starting. It is good practice to install also an indirect preheater which will furnish hot oil from the heat of boiler water or steam after the boiler is in operation. For hot-air furnace applications, where there is no hot water or steam available for indirect heating of the fuel, electric heating must be utilized entirely if heavy oils are to be burned. Preheating methods and principles are discussed in detail in Chap. XVI.

Horizontal-rotary burners probably cover the greatest range of capacities in the entire field of oil burners. In the sizes capable of burning at least No. 5 oil, and usually No. 6 as well, the range of capacities is from about 2 to 200 g.p.h., or higher, representing 600 sq. ft. of steam radiation, 960 sq. ft. of hot-water radiation as the low limit, and at least 100 times as much as the upper limit.

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CHAPTER VIII

INDUSTRIAL OIL BURNERS

The industrial applications of fuel-oil burners are numerous and varied. In general, a rough classification of applications can be made into two groups: steam boilers and technical processes. But the technical processes are so diverse that the study of industrial burners is more a matter of investigating their use than of investigating the burners themselves. So far as burner types are concerned, however, the following are recognized:

1. Horizontal rotary.
2. Low-pressure air atomizing.
3. High-pressure air and steam atomizing.
4. Aspirating, or vacuum-pressure atomizing.
5. Mechanical-pressure atomizing.

While the general tendency in fuel utilization in industrial work is of course towards the heavy grades, Nos. 5 and 6, it will be seen that frequently the nature of the process involved precludes these heavy fuels and requires rather that a lighter grade be employed. Hence it is not possible to classify industrial burners by grade of oil used, or to generalize on the correct grade of oil for industrial work, or to establish that the use of a certain grade qualifies a burner as industrial. In process work, each case must be analyzed according to its peculiarities to determine what oil to use; the type of burner is not always the determining factor, but rather is the application itself the predominant consideration.

Horizontal-rotary oil burners have been fully discussed in Chap. VII. So far as industrial work is concerned, the burner is essentially the same as for commercial work. It finds its greatest usefulness in boiler work. Such other applications as it may have will be brought out in the latter part of this chapter. The horizontal rotary is used extensively in large boiler work where the chief problems are those of control, fuel handling and furnace design. All of these topics are discussed at length in

other portions of this book. Chapter XV is particularly detailed in its discussion of the horizontal rotary in power plants, along with other types of burners.

LOW-PRESSURE AIR-ATOMIZING BURNERS

Perhaps the most versatile group of burners in the entire field of oil burning is the low-pressure air-atomizing type. Simple in design and construction, free of complicated mechanisms or intricate working parts, offering rugged performance without necessitating highly skilled attention or expensive maintenance, capable of easy installation and suitable to a host of diverse uses—these are the reasons that low-pressure air nozzles are adapted to dozens of process-heat applications.

These nozzles are made in a great variety of styles and shapes, but actually there is but little difference between the working principles—the chief significance of style being in the nature of the flame application rather than in the atomizing of the fuel. Thus some burners will have but one point of contact between oil and atomizing air, while others will have a multiplicity of such points of contact, striving thereby to gain additional advantage in effectiveness of atomization or in accuracy of control, etc. The usual fire from such burners is conical, with the slope of the fire depending on the shape of the burner orifice, but sometimes the orifice is designed to yield flat flames to suit some particular furnace requirement. Flames other than conical are not desirable unless the application demands it, for in such nonconical fires the proper mixing of oil and air is disturbed, and an inefficient combustion may result, as well as smoking and carbon formation.

There are two general types of low-pressure air atomizers—inside mixing and outside mixing—with the former predominating. The inside-mixing type lends itself more readily to fine points of design that permit close control of operation, as regards rate of oil combustion. The outside-mixing type has the advantage of greater simplicity and greater freedom from clogging and better access for cleaning.

Air pressures for low-pressure air nozzles vary from a few ounces to a maximum of about two pounds or slightly higher. The capacity of any one nozzle is proportional to the air pressure, increasing with the increase in pressure, as shown by the accom-

panying tables of capacities of typical manufacturers' products. Except in a few cases of special design, the pressure at which oil



FIG. 61.—Low-pressure air inside-mixing-type oil-atomizing nozzle. (*Courtesy of Hauck Manufacturing Company.*)

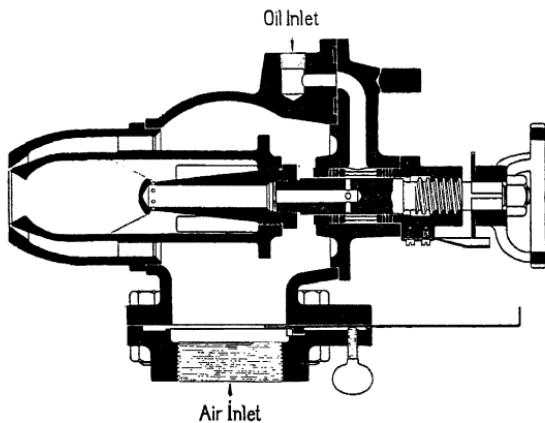


FIG. 62.—Low-pressure air inside-mixing type oil-atomizing nozzle. (*Courtesy of Schutte and Koerting Company.*)

is delivered to the nozzles is unimportant, so that the oil pumps are operated at just the required pressure to maintain the required rate of flow against the resistance of the system.

Figures 61 and 62 show in cross section two typical low-pressure air-atomizing oil burners of the inside-mixing type. Oil is fed through suitable regulating valves into the central bore, from which it issues through small slots or orifices into the main mixing chamber, the oil ports being usually at right angles to the direction of the air stream, which thereby causes the first agitating together of air and oil to form a mist. This mixture moves onward toward the outlet, where the cross section is

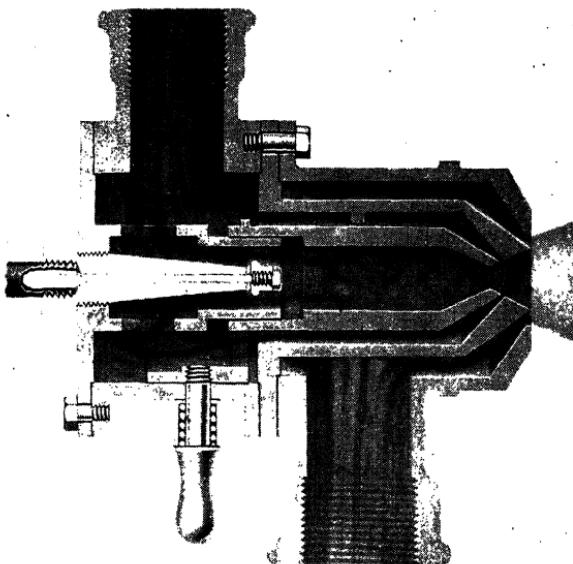


FIG. 63.—Hauck venturi triple-atomizing low-pressure air nozzle.

usually diminished by the venturi throat, causing an increase in velocity and diminution of pressure. As this occurs another stream of air strikes the mixture, completing the atomization and mixture with primary air. Ignition occurs a few inches beyond the nozzle outlet, and additional air is supplied as required to complete combustion, usually as secondary air by natural draft.

In some forms of this type of equipment the burner is constructed in such manner that the center bore, which conveys the oil and introduces it into the air stream, can be moved back and forth within a range that permits of altering its position so that

the shape of the fire can be changed within certain limits. In some special cases, the air within the nozzle is divided into three

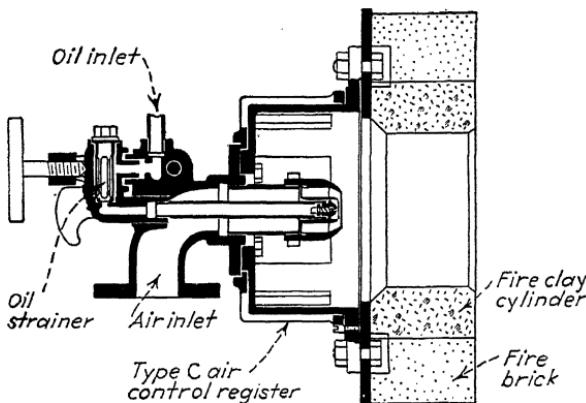


FIG. 64.—Schutte and Koerting low-pressure outside-mixing-type air nozzle.

streams, introduced at various points, to further enhance the degree of atomization or thoroughness of mixture of the fuel with primary air, as in Fig. 63.

Figures 64 and 65 illustrate two outside-mixing-type nozzles. There is but one point of contact of air with oil, and the two streams are brought together at the very tip of the burner. This type is also designated as the impact atomizer, since atomization is accomplished chiefly by virtue of the force with which the air strikes the oil—there being little opportunity for the two to be agitated and mixed together within a chamber, unless a register be provided, as in Fig. 64.

Properly designed low-pressure air nozzles can handle No. 5 oil at room temperatures, provided that the viscosity does not exceed 300 Saybolt universal sec. (at 100°F.), and that the air pressure is at least $1\frac{1}{2}$ or 2 lb. Many types can handle somewhat higher viscosities when the oil is preheated, but very often the nature of the application or the nature of the plant employing the burners is such as to preclude preheating. On the other

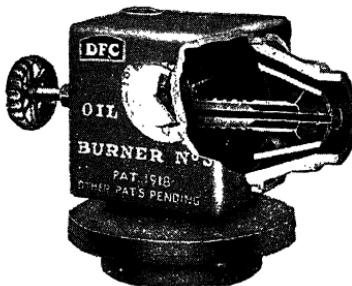


FIG. 65.—Outside-mixing-type low-pressure air atomizer. (Courtesy of The Denver Fire Clay Company.)

hand, frequently the application is of such delicate nature that the presence of smoke or fumes from improper atomization or combustion cannot be tolerated, and in such cases lighter grades of fuel are used.

Ranges of capacity of nozzles, effects of variation of air pressure, and other data on these burners are given in the tables on pp. 147, 148, and 149.

HIGH-PRESSURE AIR- OR STEAM-ATOMIZING BURNERS

These two media, high-pressure air and high-pressure steam, are so similar in their atomizing characteristics that they are

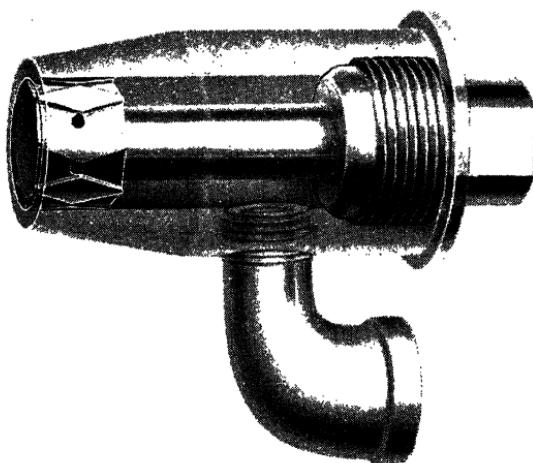


FIG. 66.—Hauck inside-mixing high-pressure air- or steam-atomizing burner nozzle.

interchangeable in the same equipment, and each is as effective as the other regardless of burner design. It is imperative, however, that steam used for oil atomization shall be dry; slugs of water or entrained moisture have a very harmful effect in oil atomization.

As in the case of low-pressure air atomizers, high-pressure air or steam atomizers can be considered as of two types; inside mixing and outside mixing; and again the inside-mixing type is the more commonly used. The inside-mixing burner with high-pressure atomizing medium is somewhat simpler in construction than in the case of low pressure and there are fewer parts. The

TABLE XV.—AIR VOLUME, OIL-BURNING CAPACITY AND PIPE SIZES FOR DENVER FIRE-CLAY LOW-PRESSURE BURNERS
AT VARIOUS AIR PRESSURES

Inlet size, in.		Air pressure																	
		6 oz.				16 oz.				24 oz.				32 oz.					
Burner no.	Air pipe	Oil pipe		Air, c.f.m.		Oil, g.p.h.													
		Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.
1	2	1½	½	67	13	2.8	0.5	109	21	4.7	0.9	133	25	5.7	1.1	155	29	6.6	1.3
2	2½	1½	½	80	13	3.4	0.5	130	21	5.6	0.9	160	25	6.9	1.1	185	29	8.0	1.3
3	3	1½	½	133	24	5.7	1.0	217	39	9.3	1.7	266	47	11	2.1	308	55	13	2.4
4	4	1½	½	320	101	14	4.3	495	164	21	7.0	608	201	27	8.6	707	233	30	10
5	5	1½	½	655	165	28	7.0	1068	268	46	12	1310	330	56	14	1510	381	65	16

OIL BURNERS

TABLE XVI.—AIR VOLUME, OIL-BURNING CAPACITY AND PIPE SIZES FOR HAUCK LOW-PRESSURE BURNERS
AT VARIOUS AIR PRESSURES

Burner no.	Inlet size, in.	Air pressure						32 oz.					
		8 oz.			12 oz.			16 oz.			24 oz.		
		Air, c.f.m., max.	Oil, g.p.h.	Air, c.f.m., max.	Air, g.p.h.	Oil, g.p.h.	Air, c.f.m., max.	Air, c.f.m., max.	Oil, g.p.h.	Air, c.f.m., max.	Air, c.f.m., max.	Oil, g.p.h.	Air, c.f.m., max.
578	1	10½	1	½	13	1½	½	15	1½	½	18½	½	2½
580A	1½	24½	2½	1	30	3	1	34½	3½	1	42	4	5½
581	2	71	7	2	92	9	2	101	10	2	124	1	1
582	3	156	15	3	202	20	3	221	22	3	271	2	2
583	4	193	19	6	250	25	6	274	28	6	336	3	3
584	6	535	53	8	685	68	8	750	75	8	915	8	8
												1060	106

TABLE XVII.—AIR VOLUME, OIL-BURNING CAPACITY AND PIPE SIZES FOR HAUCK TRIPLE-ATOMIZING Low-PRESSURE BURNERS
AT VARIOUS AIR PRESSURES

Burner no.	Inlet sizes, in.		Air pressure													
			8 oz.				16 oz.				24 oz.				32 oz.	
	Primary and secondary air	Tertiary air	Oil	Air, c.f.m., max.	Air, g.p.h., min.											
315	1 1/4	1 1/2	1/4	54	3 3/4	4	78	3 1/4	8	97	1	10	114	1 1/2	14	
316	2	2 1/2	3/8	230	1 1/2	12	330	2	25	400	3	35	460	5	50	
317	2	3	3/8	350	2 1/2	20	500	4	40	605	5	50	700	7	70	
318	2 1/2	4	3/8	480	3 1/2	32	685	6	55	831	8	70	960	10	90	

fuel is broken up principally by the expansion of the atomizing fluid. The pressure under which oil is delivered is again unimportant, and need not be higher than that required to effect entrance through the oil port. As the atomizing medium is released from its confining pipe before the oil is brought into contact with it, the oil does not have to be pumped against the pressure at which the atomizing fluid is delivered.

When steam is used for atomization, it is important that the burner design be such that the amount of steam required be held to a minimum, as steam is generally expensive, besides representing a loss of distilled water from the boiler feed. In marine

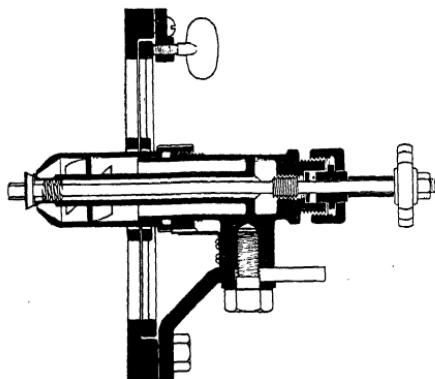


FIG. 67.—High-pressure air or steam outside-mixing atomizer. (*Courtesy of Schutte and Koerting Company.*)

practice the use of steam has been discontinued entirely because of the water loss. In stationary practice, steam still has its field of usefulness, principally because it makes possible the use of extremely heavy fuels, such as gashouse tar, but even in stationary boilers the consumption of steam for atomization should be held below 2 per cent of the steam generated. Older types of steam atomizers which used as much as 3 to 5 per cent of the generated steam for atomizing are now quite obsolete, and their use cannot be countenanced by modern oil-burner practice. Steam nozzles have to their advantage, besides the ability to break up very heavy fuels as already mentioned, the facts that the ordinary grades of fuel do not require the temperature of preheating as high as for other burners, and that the initial cost of the equipment is low, since heavy pumping equipment is not needed to maintain high oil pressures.

High-pressure air nozzles can be considered, usually, only when there is already available in the plant a source of high-pressure air, for the installation and operation of compressors for the sole purpose of running oil burners is too expensive. They can be successfully applied to the heaviest grades of fuel if adequate preheating is provided, although in common with steam nozzles they have the fault of being extremely sensitive to slight changes in pressure and, therefore, of requiring close attention.

A typical high-pressure air or steam nozzle of the inside-mixing type is shown in section in Fig. 66. After the oil leaves the oil tube it is picked up by the atomizing medium and broken up by

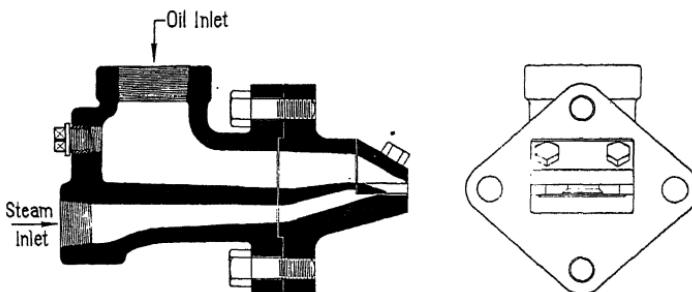


FIG. 68.—Schutte and Koerting flat-flame outside-mixing high-pressure air or steam nozzle.

the expansion occurring within the mixing chamber. If the outlet from the oil tube be constricted and some pressure be applied to the oil, preliminary atomizing effect is imparted to the oil before it enters the mixing chamber.

The outside-mixing type can be similar to that for low pressure, as shown in Fig. 65, or can be made as shown in Fig. 67. Here the steam or air expands outside the nozzle, and in so doing picks up and breaks up the liquid fuel. In Fig. 68 is shown an outside-mixing type which produces a flat flame.

Since steam requirements for atomization have already been stated at 2 per cent maximum, this establishes a ratio of about 2.5 lb. of steam per gal. of oil. When high-pressure air is used, the required weight is somewhat less, running to 1 to 1.5 lb. of air, or 15 to 20 cu. ft. of free air, per gal. oil. The minimum operating pressure for either air or steam is 40 lb. per sq. in.; the usual maximum pressures of these two atomizing media are around 80 lb.

ASPIRATING OR VACUUM-PRESSURE ATOMIZING

This burner depends for its operation upon the suction created by a stream of air in passing over the open end of a tubing, commonly known as the aspirator effect. Since the vacuum which lifts the oil up into the air stream is created by air blown under pressure, the name vacuum-pressure has also been used to designate this type of burner. In practice, air pressure up to 5 lb. in most types, but as high as 15 lb. in some, is developed by a slow-speed rotary air compressor. This air is directed over the open end of a tube which connects into the oil reservoir. The amount of oil drawn up into the air stream will vary directly with the pressure of the air, so that direct proportioning of oil and primary air can be secured merely by regulating the air pressure delivered through the nozzle. The action of air stream on the oil picked up breaks the oil into finely divided particles and yields a combustible mixture. Secondary air is admitted through properly located air ports, and is drawn into the firebox by natural draft. Normally, about 5 per cent of the total air is primary and about 95 per cent is secondary. The burner will handle grades of oil as heavy as No. 4, without preheating, and although heavier oils can be atomized if preheated, it is not common practice to use this equipment where the use of heavier fuels is required.

At an air pressure of 5 lb., the maximum capacity per nozzle is 3 gal. per hour; as the atomizing air pressure is raised up to 15 lb. per sq. in., the capacity of a single tip can be increased up to 10 or 12 g.p.h. Commercial forms of this equipment permit of combining up to three tips into a single frame or block, and if required, two or three blocks can be operated from a single air compressor.

Since oil is aspirated from the reservoir (Fig. 69) beneath the nozzle, it is necessary that the oil level be maintained uniform. This is accomplished by a float valve if the oil be fed by gravity or by a float switch if the oil is delivered from storage by a pump.

For automatic operation, ignition is accomplished by any of the three standard methods: expanding gas, electric-gas or electric spark. These burners can also be readily adapted to high-low operation by causing a pressure-relief valve on the air pipe to be opened at such times as low fire is required. Modulating control could also be secured if means be provided to gradually increase

and decrease the air pressure. It is merely necessary to control the air stream—oil feed automatically proportions itself to the air pressure. Of course, under variable fire operation, it is neces-

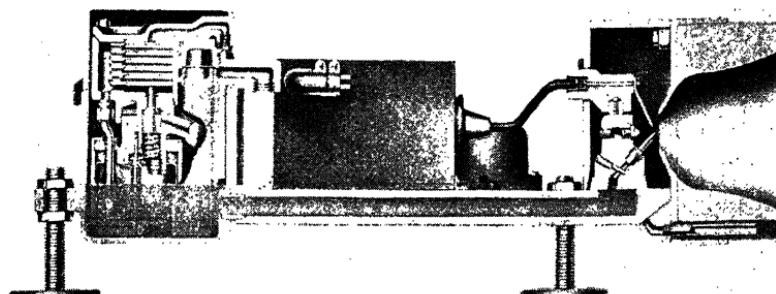


FIG. 69.—Aspirating or vacuum-pressure burner. (*Courtesy of Caloroil Burner Corporation.*)

sary to regulate secondary air as well by synchronizing the movement of the secondary-air damper with the setting of the primary-air pressure.

MECHANICAL PRESSURE-ATOMIZING BURNER

The mechanical pressure-atomizing oil burner is used almost exclusively for boiler-plant work, and most major-sized boiler plants, whether industrial or central station, will employ this type of burner for burning fuel oil. The nature of the equipment, especially the auxiliaries such as preheaters, makes it most adaptable to plants which generate at least 100 lb. steam pressure, and which have capacities upward of 250 or 300 boiler hp.

In the mechanical pressure-atomizing burner, oil is atomized by pump pressure in combination with a properly designed atomizing tip. The nozzle itself consists of a burner tip, built of high-temperature resistant metal (usually alloy steel), through which is drilled a hole proportioned to the quantity of oil to be burned, and of an atomizing disk, consisting of a steel disk into which are cut a series of tangential slots. Oil passing through the slots is given a whirling motion, then passes through the turbulence chamber between the disk and tip and finally emerges through the tip which completes the atomization. Usually, the atomizing disk has attached to it a steel cylinder with drilled openings to serve as a final strainer before the oil enters the atomizing head.

The burner atomizer and strainer are supported in an alloy-steel cylindrical housing, onto the outer end of which is screwed a nut that holds the tip in position, and onto the inner end of which is screwed the oil-feed tube. The entire assembly fits into a jacket tube, arranged with a wing screw so that the burner itself can be removed from the jacket quickly and without the use of special tools. The wing screw clamps down against a special union bushing, the junction through which oil enters into the oil-feed tube. Oil is brought up to this point through a circulating pumping and piping system. The reason for having the wing screw device for quickly changing burners is that the heavy oils burned usually require that clean burners be installed at about

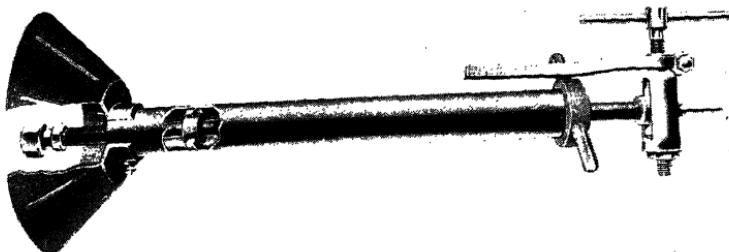


FIG. 70.—Bethlehem-Dahl high-pressure mechanical atomizing nozzle assembly.

8-hr. intervals. An ample number of spare burners must be provided.

With any atomizing disk-and-tip combination, a certain amount of fire regulation can be secured by varying the oil pressure within definite limits. The minimum operating oil pressure is 75 lb.; the usual maximum is about 200 lb., although in special cases higher pressures may be used, especially in marine work. In general, the minimum gallons per hour at 75 lb. atomizing oil pressure, on any given disk-and-tip combination, is about 50 per cent of the maximum at 200 lb. pressure. Hence, whenever the minimum fire at 75 lb. pressure is still too large for the load, a smaller tip must be inserted, or if there be more than one burner per boiler, one or more burners may be shut off completely. Similarly, whenever the capacity at 200 lb. oil pressure does not yield an adequately large fire, a large tip must be inserted.

If, however, a particular plant has continuously wide fluctuations in steam demand and if, in addition, automatic control is

desired, then a different arrangement is employed. In such cases, a twin-plex or double-nozzle burner is used. This burner differs from the standard in that it has two separate atomizing chambers, which supply a common tip. Two sets of slots, having, respectively, low and high capacities, are cut into the disk. The smaller atomizer is designed for the minimum practical fire, while the larger, of course, is sized for the probable maximum. Thus, by means of a system of valves and levers, usually actuated automatically, the high side is throttled down until it is

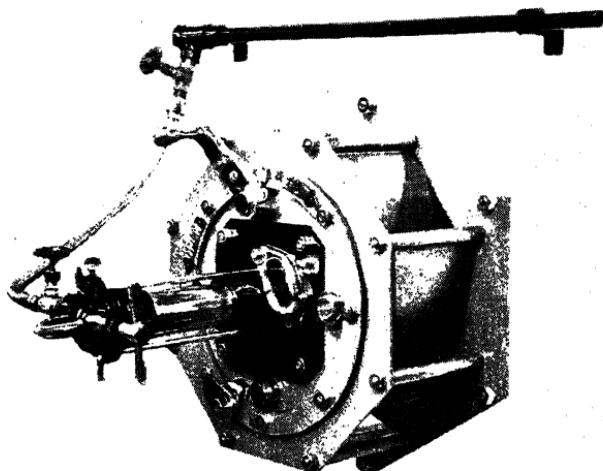


FIG. 71.—Todd "Hex-Press" mechanical pressure-atomizing burner with air register.

completely closed, leaving the low side only in use. The low side can then be throttled down also, but only to its minimum fire. A by-pass valve is usually installed to allow the minimum quantity of oil to pass through at 75 lb. pressure, regardless of the position of the low-side regulating valve. Control can be effected by closing of valves at the burners or, in the case of the standard single-nozzle burner, by adjusting oil pressure at the pumps. In construction, the twin-plex burners consist of two concentric oil-feed tubes. Oil through the center tube feeds the larger nozzle; oil between the two tubes feeds the smaller.

All air for combustion comes through the air register which is always part of this type of burner installation, and air flow is

OIL BURNERS

regulated by setting of louvers or other devices provided for this purpose in the register. Whenever natural draft is inadequate for the needs of the installation, forced draft can be provided, and air under pressure taken through the register, subject to the same type of regulation as with natural draft.

The position of the burner with respect to the boiler front (Fig. 71) can be varied by moving it forward and backward,

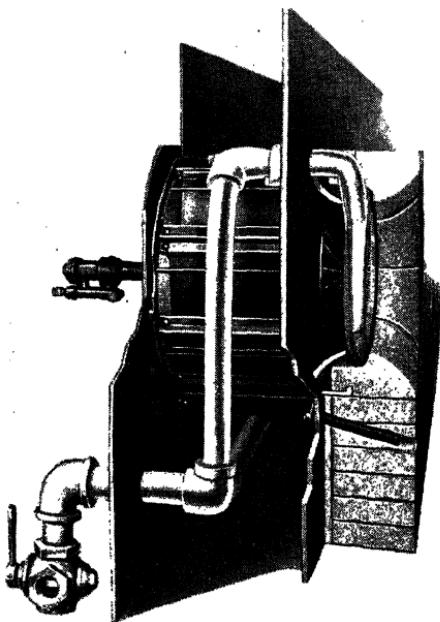


FIG. 72.—Mechanical pressure-atomizing burner mounted for forced draft operation, with combination gas-burner ring. (*Courtesy of Babcock and Wilcox Company.*)

thereby also changing its relative position with respect to the burner front and venturi tube. The apparatus which carries the burner, allowing for the forward and backward movement, is called the carrier tube, and at its end is a flame cone, which assists in holding the fire near the end of the burner and also helps in preventing cold incoming air from striking directly into the oil spray.

When designed for forced draft, the burner is set between two steel plates which form a wind box. The inner plate (Fig. 72),

closest to the boiler, supports the burner in position. The air supply, usually delivered from a blower, enters through the bottom of the wind box, sometimes first traversing under the furnace floor or through air-cooled walls. Besides cooling the floor or walls, this also preheats the air and thus assists in securing complete combustion. Blowers intended for this use should have a capacity to deliver the entire air supply at maximum oil-firing rate, at the proper static pressure to deliver the required amount of air through the registers. When the air is heated to fairly high temperature the air register should be insulated.

In both the natural-draft and forced-draft mountings, the entire burner assembly can be swung away from the furnace front by means of hinged supports. It is necessary, of course, that the oil be brought up to the burner feed tube through a flexible oil conduit.

Fuel oil for use in mechanical pressure-atomizing burners must always be preheated, but when adequate preheating is provided, the heaviest oils can be burned. Whatever may be the oil's viscosity at room temperature, the temperature of the oil must be raised to that point at which the viscosity is reduced to at least 200 sec. Saybolt universal.

The maximum oil-burning capacity of a single burner, with natural draft, is about 125 g.p.h. With forced draft, the maximum is about 200 g.p.h. With either natural or forced draft, the minimum fire that can be successfully carried is around 15 g.p.h., although, of course, the 15 gal. minimum could not be obtained on a burner having the maximum capacity mentioned. To pass down through the full range, it is necessary to change burner tips.

Mechanical pressure-atomizing burners have these advantages over other types of oil burners:

1. On natural draft, no blowers or air compressors or motors are required for accomplishing atomization (aside from the oil pump, which is needed on all other types as well).
2. Because oil pressure only is needed, operating costs involve no such items as atomizing steam, air or electricity.
3. Because oil pumps can be driven either from steam or electric motors, the plant is independent to the extent that it can use either, whenever the other is not available.
4. Maintenance costs are less, especially when more than one burner is required per boiler or where more than one boiler is

used. This is because of the complete absence of shafts, bearing, motors, turbines or other driving mechanisms. There are no moving parts on the pressure-atomizing burner.

5. Per developed boiler horsepower, the burner mechanism costs less, although in plants having elaborate control systems, the auxiliary apparatus will cost more than for other types of burners.

6. These burners are extremely quiet in operation.

Mechanical-atomizing burners have these disadvantages, against other types of burners:

1. Frequent cleaning of the burner tips and disks is necessary.
2. Range of capacity by varying pressure is held within close limits.

COMBINATION GAS AND OIL BURNER

It is possible to adapt the mechanical pressure-atomizing burner to the use of a second fuel, either gas or pulverized coal. The most frequent adaptation is the combination gas-oil burner. The same type of air register is used, as is also the same type of oil burner, and a gas ring, Fig. 72, is fitted into the common assembly, in a position roughly encircling the burner tip. The ring is drilled with the required number of gas ports of suitable diameter and consists, essentially, of an annular casting mounted upon the same hinged supports so that it also can be swung out of the firing position for inspection and cleaning.

The general shape and location of the gas ring cause entering air, by either natural or forced draft, to swirl through the space between the outside of the ring and the register, and between the inner surface of the ring and the flame cone, so as to cause a turbulence of air and gas and thereby bring about an intimate mixing, as well as to partially induce a suction of gas into the furnace. The gas ring can also be caused to move forward and backward, changing its position with respect to the furnace wall, to permit of securing the most advantageous alignment, and of altering the shape and relationship of the flame with respect to the setting.

The two fuels, gas and oil, may be burned separately or simultaneously. In general, the same control apparatus used for oil regulation can be utilized to control the flow of gas and the size of the gas flame.

In some cases and types of equipment, pulverized coal has been combined with the pressure-atomizing burner in much the same way as with gas. It is not, however, a generally successful method of burning pulverized coal. This latter fuel apparently is best handled in equipment which is especially designed for its use and application.

INDUSTRIAL OIL-BURNER APPLICATIONS

The actual installation of oil burners, including all mechanical operations such as oil pumping and piping, firebox construction, electric controls and wiring, are discussed in other chapters in this book. The complete application of oil burners to stationary boilers is covered in some detail in Chap. XV. The remainder of this chapter will be devoted to a discussion of general features of applying oil burners to the great variety of technical processes which constitute a large part of the field of usefulness of liquid fuels.

Locomotives.—In steam locomotives the boiler firebox is at the cab (the tubes of the fire-tube boiler running forward), there being but one pass. By the term front of the firebox, however, is meant the cab end; the tube sheet end is called the rear of the firebox. Most oil-burning locomotives are rear fired, burner or burners being either at the extreme end of the firebox, under the tube sheet, or else at a point under the firebox near the end. Steam is the universal atomizing medium in locomotive work. This means that once the engine leaves the roundhouse, it is imperative that steam pressure be maintained at all times, even though the engine be idling. For this reason the oil-feed valve or cock is usually provided with a by-pass which assures a minimum flow that will maintain a fire even though the fireman should inadvertently close down the flow when his load drops off. Sometimes the same end is accomplished by having a small burner, called the pilot burner, which runs all the time from the moment the fires are started in the roundhouse. All regulation beyond the fire of the pilot burner is done by varying the flow to the main burner, which can be shut off entirely.

Figure 73 illustrates a typical locomotive setting of nozzles and refractory. Air for combustion is generally admitted through ports located at some distance from the nozzles, and frequently a boxlike arrangement is used, having two dampers, so that air

may be drawn in through that side toward the direction of travel. In the cab are located control valves and levers so that the fireman can regulate his firing rate and air admission as desired.

Oil pumps are not used in locomotive work. The oil flows to the nozzles by gravity there being sufficient head in the height of the fuel tank over the nozzles to establish the flow. Heating coils are contained within the tank, and another heater is placed in the feed line.

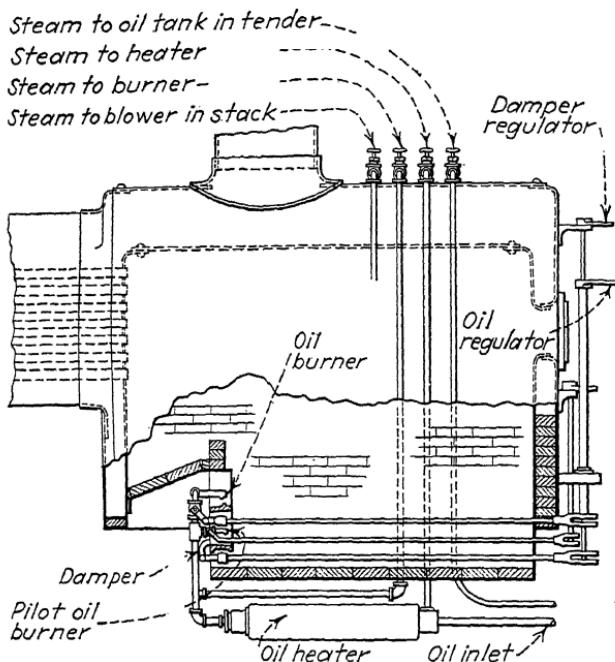


FIG. 73.—Typical locomotive oil-burner setting.

Marine Boilers.—The opening years of the twentieth century saw the use of fuel oil firmly established as a fuel for marine boilers. As early as 1887, Schutte had applied for a patent on an oil-burner design especially for steamship application. During the period 1901–1904, the United States Navy conducted exhaustive tests on solid and liquid fuels, and as a result adopted oil as the standard fuel for the Navy. Within a few years fuel oil became the prevailing fuel for all ocean vessels. Marine practice in general conforms to accepted stationary boiler practice—whatever difference exists lies rather in the boiler design than

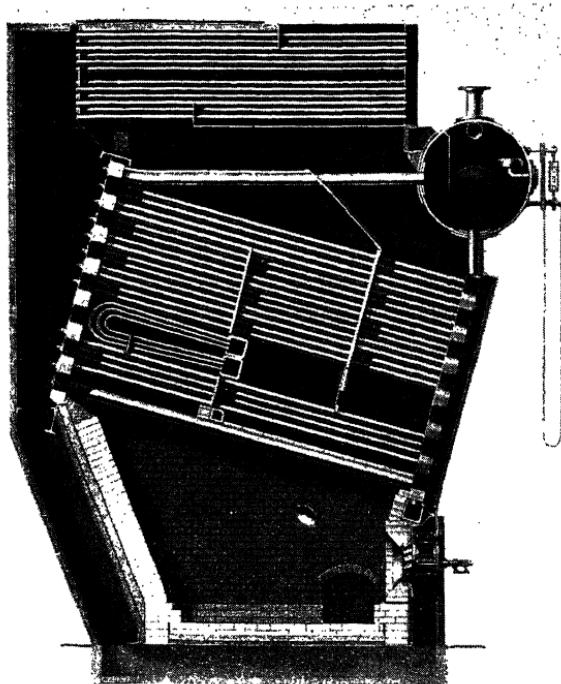


FIG. 74.—Typical marine-boiler setting, mechanical burners with forced draft.
(Courtesy of Babcock and Wilcox Company.)

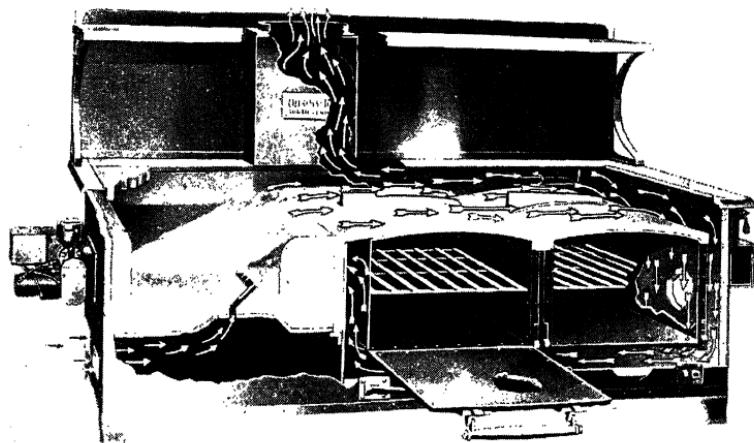


FIG. 75.—Oil-fired restaurant range. *(Courtesy of South Bend Range Company.)*

in the burner design. Full treatment of large boiler-plant operation is given in Chap. XV, but a typical marine-boiler setting is shown in Fig. 74.

Food Processes.—Oil burners are used in a great variety of food industries, practically for every heat application, whether roasting (Fig. 75) cooking, baking, boiling or sterilizing. Included in the long list of products prepared by oil-fired processes are: meat products, bread, crackers and pastries, chocolate and candies, sirups and preserves, cereals and salt. The ease of temperature control makes oil an especially desirable fuel for this work, as well as the absence of attention incident to oil-fuel firing, leaving the attendant free to watch the process itself.

To illustrate the general principles of oil-burner application to food processes, a detailed study is presented of bake oven operation. It will be found that the methods discussed, together with the underlying theory and reasons for them, will be applicable throughout this field.

There are four broad classifications of bake ovens:

1. Peel.
2. Traveling.
 - a. Straight traveling.
 - b. Tray.
3. Rotary.
4. Ferris wheel.

The peel is the oldest type, and is made of two forms—brickset and portable. Essentially, the peel oven is simply a stationary refractory hearth within a refractory enclosure, the brickset being constructed with a permanent masonry setting of somewhat massive construction, while the portable is so made that it may be knocked down and rebuilt at any desired location. Peel ovens may be either direct or indirect fired, meaning that the combustion gases may pass directly through the baking chamber, as in direct firing, or may pass underneath and then around the baking chamber, as in indirect firing. Either type usually operates on natural draft.

The first type of traveling oven in common use had a movable hearth which traveled in a horizontal plane onto which the bakery goods were introduced at one end to emerge completely baked at the other. This straight traveler is always indirect fired, usually from three or more points. The second type of traveling oven,

designated as the tray oven, has trays set on a conveyer system, so that the product to be baked, after being placed upon trays, is carried back and forth within the oven enclosure, but emerges at the same end as introduced. These may be either direct or indirect fired, but in both cases are forced draft, the gases of combustion being circulated and forced through the oven by a fan. Part of the gases is recirculated after the first circuit



FIG. 76.—Oil-fired traveling bake oven with six furnaces. (*Courtesy of Baker Perkins Oven Company and Simplex Oil Heating Corporation.*)

through the oven, being returned to the beginning to merge with fresh combustion gases from the firebox, while the remainder is spilled to the stack.

The term rotary oven is applied to a single- or double-decked round horizontal hearth that is rotated by a vertical motor-driven shaft at the oven's center, the entire assembly being enclosed with either a portable or fixed setting, the former predominating. These ovens are smaller than the traveling type, and are widely used. They are always direct fired, under natural draft.

The ferris-wheel oven might be termed a vertical-tray type, except that such a name would confuse it with the traveling-tray oven. The ferris-wheel oven is built just as its name implies, of a series of trays carried at the perimeter of a vertical revolving wheel, the trays being freely supported to retain a level position throughout the revolution of the wheel. In recent years the oven

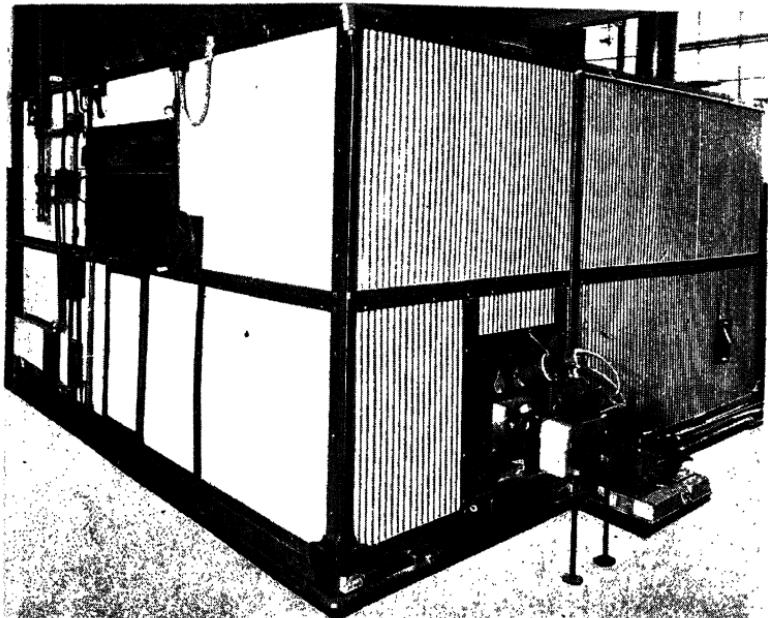


FIG. 77.—Portable rotary oven with aspirating burner. (*Courtesy of Middleby-Marshall Company and Mid-West Heat Service Company.*)

has had wide application, particularly in smaller bake shops. It may be either direct or indirect fired, under natural draft.

The domestic pressure-atomizing burner was the first burner utilized for bakeries. In many cases it worked out very well; in others the results were inferior or completely unsatisfactory. Its chief success was on peel, smaller type tray and the ferris-wheel ovens, all indirect, in capacities ranging from 2 to 15 gal. of oil per hr., and burning No. 3 or lighter. On peel ovens, the burner is best operated manually, being started by hand and allowed to run until the baking chamber is at the required maximum temperature. As the oven slowly cools, various kinds of bakery goods are sent in at the optimum temperature for each. At the

desired minimum point, the burner must again be started, to repeat the cycle as often as necessary. The usual range of baking-chamber temperature is from 300 to 600°F. It is essential that this burner be mounted in such fashion that it may be pulled away from firing position when not in use, to minimize the ill effects of reflected heat. When so pulled back, the burner should be protected from the furnace heat by a shield placed over the furnace front. With all these precautions, the pressure-atomizing nozzle still has the inherent objections that the high combustion-chamber temperatures cause coking of oil within the tip, and

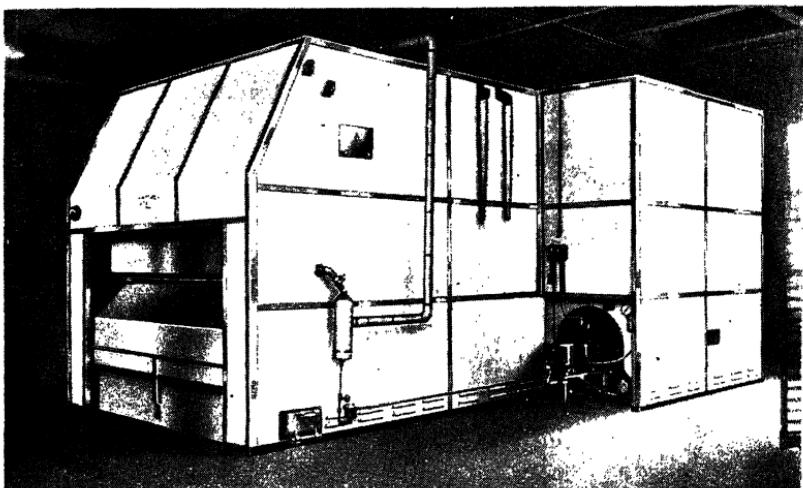


FIG. 78.—Recirculating tray oven, indirect fired by aspirating burners.
(Courtesy of Petersen Oven Company and Mid-West Heat Service Company.)

also damage the electrodes and porcelains which are used for electric ignition—as applied to peel ovens.

In the case of small tray ovens, automatic control may be used, the burner operation being intermittent under the control of an oven thermostat. The high air velocities resulting from the circulating fan assist in cooling the burner tip and electrodes, so that the reflected heat effects are not so marked as with the peel installation. Ferris-wheel ovens are natural draft, but as a rule the refractory is not so massive as with peels, so again reflected heat is not so intense. Automatic intermittent operation is also customary in the latter case. Most oven thermostats, used in conjunction with pressure-atomizing or other automatic burners, will give temperature regulation within 5°F.

High- or low-pressure air-atomizing and steam-atomizing burners have been applied to a considerable extent to both direct and indirect peel ovens, as well as to some older types of straight traveling ovens. Various fuels can be used, varying all the way from Nos. 1 to 6. On the West coast particularly, the low-pressure air nozzle is widely used, in conjunction with both direct- and indirect-fired peels. On the direct fired, No. 3 is about the heaviest oil that can be safely used. There has been some appli-

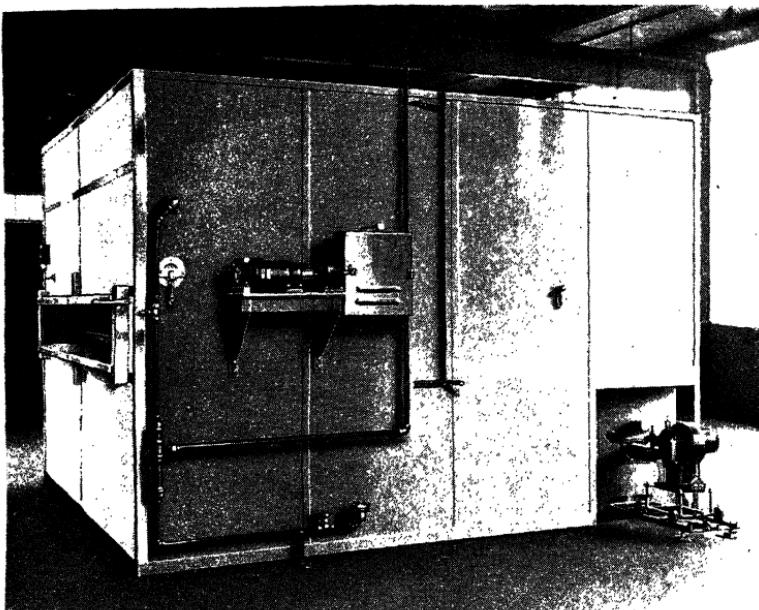


FIG. 79.—Petersen ferris-wheel oven indirect fired by pressure-atomizing burner.

cation, however, of heavier fuels to direct peels, with high-pressure air and steam nozzles, but these two burners have their greatest application to multiple-chamber traveling ovens, usually with Nos. 5 and 6 oil preheated. High-pressure air and steam nozzles do not lend themselves well to even semiautomatic control, and intermittent automatic operation is not possible at all. Since they must be manually operated, they will be found mostly in larger bakeries, on rotary, ferris-wheel and traveler ovens.

Horizontal rotaries can be universally employed on all types of indirect ovens, under either manual or automatic operation, but are not very well suited to direct firing because of a slight tend-

ency to smoke. As explained in Chap. VII, horizontal rotaries can run on almost any oil, from the lightest to the heaviest, provided that preheating is used on No. 6. As a rule, manual operation prevails on peels, and intermittent automatic on other types of oven. Ignition may be expanding gas or electric-gas. The capacity range is from 3 to 20 g.p.h. per hour.

The aspirating or vacuum-pressure burner has the most universal application to bakeries, there being practically no restrictions on its use with any direct or indirect oven, but of course there is a limitation of the grade of oil to No. 3 or lighter, except as noted. On straight traveler, tray or ferris-wheel ovens, full-automatic intermittent operation has been commonly used, with ignition by expanding gas, electric-gas or straight electric. In a few cases of manual operation on indirect ovens, preheated No. 6 oil has been used. Owing to the relatively large air and oil orifices used, clogging of tips on aspirator burners seldom occurs.

The following tabulation gives some idea of relative efficiencies of various types of bake ovens.

TABLE XVIII.—B.t.u. CONSUMPTION PER POUND OF DOUGH FOR BAKE OVENS

Indirect brickset peel.....	1200 to 2000
Indirect recirculating semiportable peel.....	800 to 1200
Indirect recirculating tray.....	500 to 660
Indirect recirculating straight traveling.....	600 to 750
Direct recirculating tray.....	450 to 550
Rotary.....	500 to 800

CERAMIC INDUSTRY

This very important and extensive branch of industry employs oil almost to the exclusion of all other fuels, although gas would be the second most popular fuel. There are in general two types of furnaces for the baking of clay or earthenware products, batch and continuous. In the former the furnace is loaded with the fabricated clay articles, then sealed up and the fires are started, slowly at first, so that the temperature may be raised at a rate best suited to the product being baked. After the maximum temperature has been reached and sustained for the required length of time, the burners are gradually cut down so as to reduce the temperature at the desired rate. When the furnace is finally down to ordinary temperatures again the furnace is broken open

and the finished products removed. In the continuous furnace a traveling platform carries the raw formed clay articles into the furnace, the point of entrance being at low temperature. Then, as the platform continues on into the furnace, the temperature is gradually raised, reaching the maximum, then lowered until at the exit point the temperature is very close to room tempera-

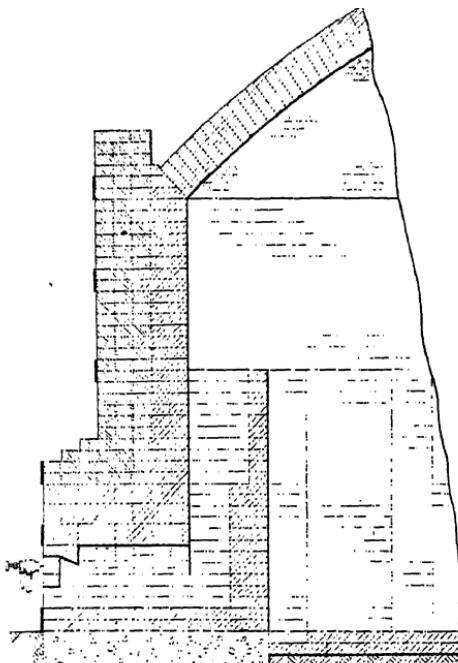


FIG. 80.—Down-draft brick kiln.

(*Fire Clay Company.*)

ture. In the batch process, the oil burners may be controlled by hand, although control by pyrometer is much to be desired. In the continuous furnace, pyrometric control is indispensable. Typical of such installations is Fig. 80, showing details on firebox construction and oil-burner arrangements for a down-draft brick kiln.

INDUSTRIAL CHEMICAL APPLICATIONS

Numerous basic chemical industries involve heat processes, and many times fuel oil can be adapted to such uses with excellent results. Among these are: glass furnaces, lime and cement kilns,

(Courtesy of *The Denver Fire Clay Company*.)

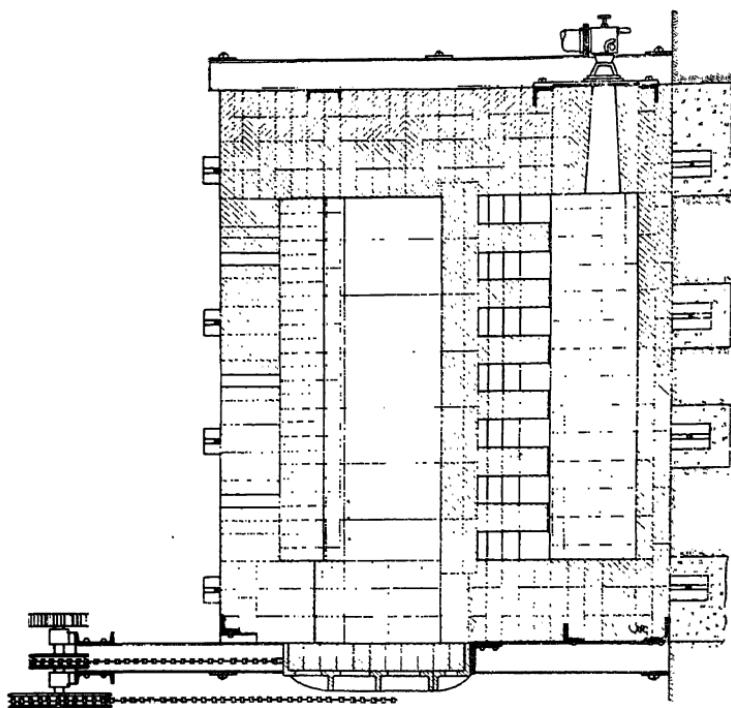
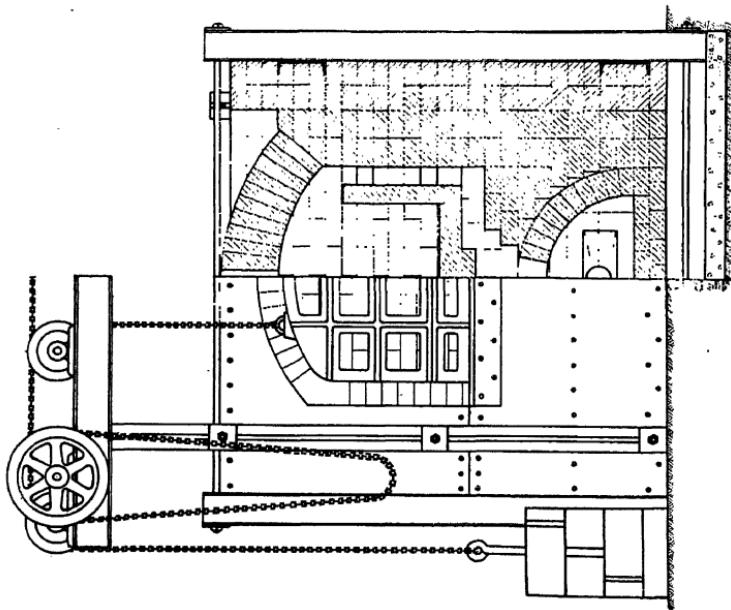


FIG. 81.—Medium-sized muffle furnace.



enameling and drying ovens. Again, temperature control is vital to successful results. The processes themselves are so numerous and diverse that space does not permit of complete description here. The illustrative material given on these pages covers typical applications.

METALLURGICAL USES

The metallurgical uses of oil burners undoubtedly embrace the longest and most diverse list of applications, including as they do: ore roasters, reverberatory furnaces, open hearths, crucibles, forges, annealing furnaces, galvanizing vats, etc. A full discussion of all these would amount to a treatise on metallurgy, and cannot be offered here. Suffice it to say that oil performs a most vital function in the operations of both ferrous and nonferrous metallurgy. The accompanying drawings and photographs give as much detail as is here permitted on the design and construction of the furnaces and installation of the burners.

TABLE XIX.—COMPARATIVE ATOMIZING COSTS BY DIFFERENT METHODS

Cost of Power at 1 ct. per Kw.-hr.

Labor and Fixed Charges Not Included

	Per 1000 Gal. Fuel Oil
Horizontal rotary.....	\$0.20
Mechanical pressure.....	0.40
Low-pressure air, $\frac{1}{2}$ lb.....	0.50
Low-pressure air, 2 lb.....	0.75
Steam*.....	1.00
Medium-pressure air, 10 lb.....	1.30
High-pressure air, 80 lb.....	4.70

* Cost of steam for atomization taken at $2\frac{1}{2}$ per cent of oil burned at 4 cts. per gal.

THERMOMETRIC AND PYROMETRIC CONTROLLERS

Industrial-type thermometers are very useful in many oil-burner applications to process industries, for both indicating and recording furnace temperatures and for regulating burner operation in accordance with furnace temperatures. Thermometers are self-contained units comprising body, tubing and bulb. The bulb is the temperature-sensitive element; the tubing connects the bulb to the body and transmits the effects of temperature from the bulb to the indicating and controlling mechanism in the body. The maximum temperature to which this equipment can be used is 1200°F.

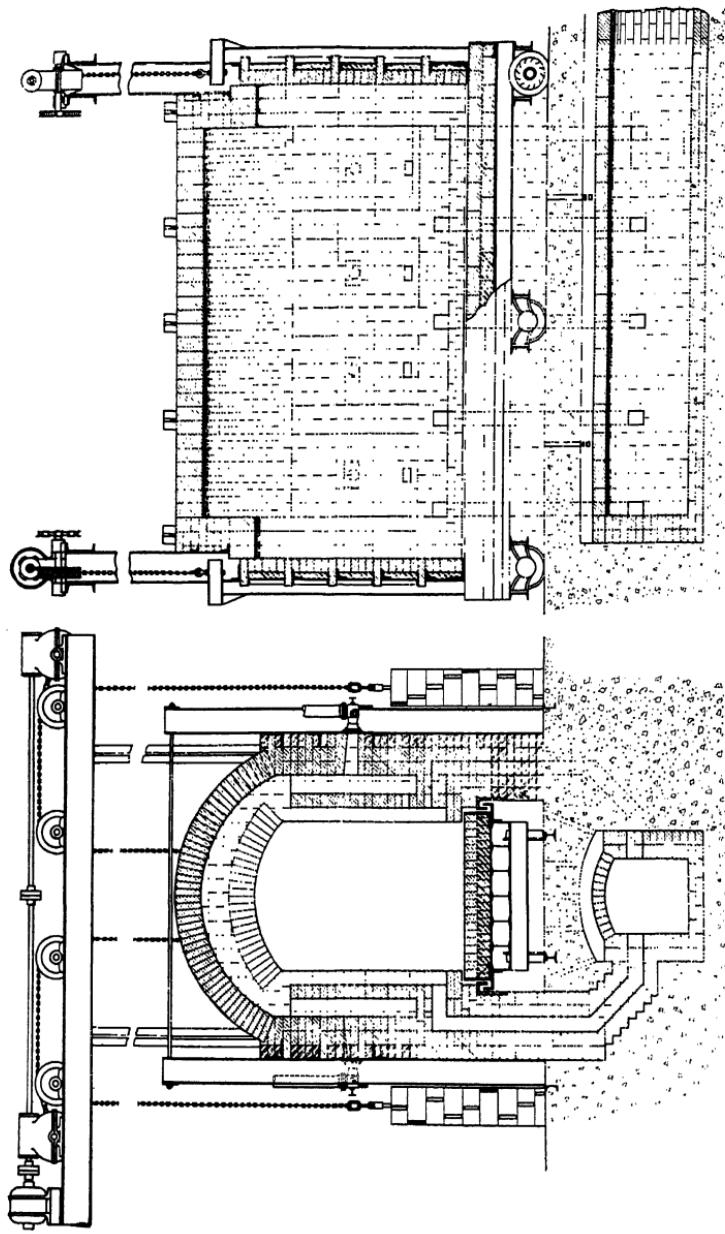


FIG. 82.—Large ear-annealing furnace.
(Courtesy of *The Denver Fire Clay Company*)

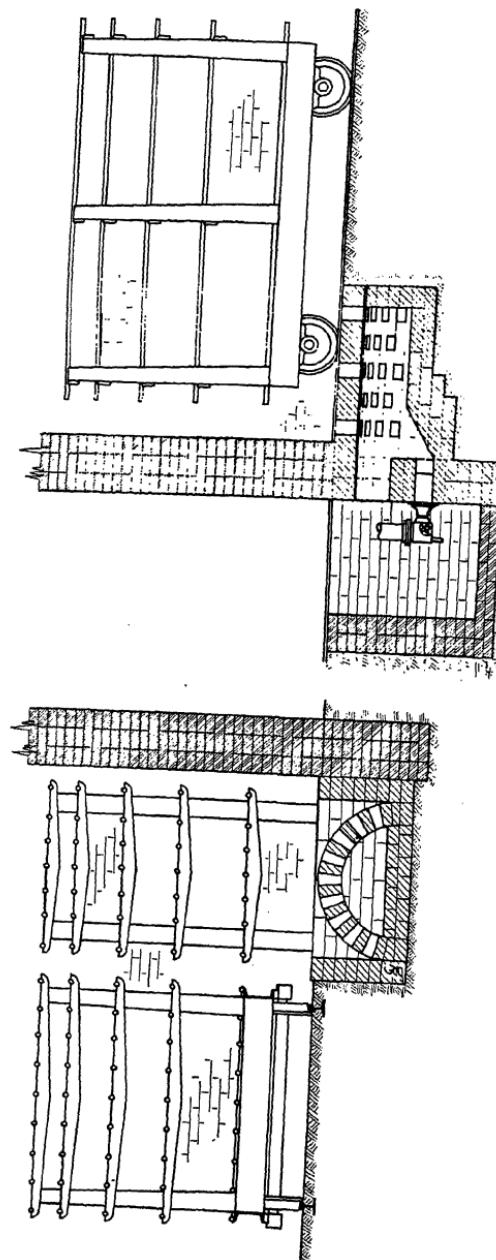


FIG. 83.—Large core oven. (*Courtesy of The Denver Fire Clay Company.*)

Bulbs are made temperature sensitive by containing either of the three classes of fluid: a permanent gas, such as nitrogen,

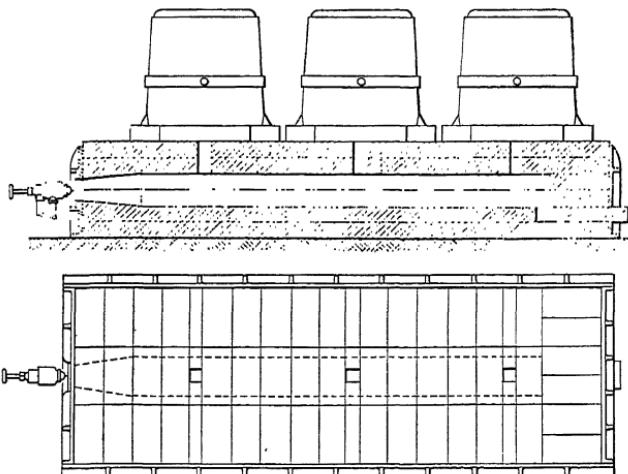


FIG. 84.—Ladle heater. (*Courtesy of The Denver Fire Clay Company.*)

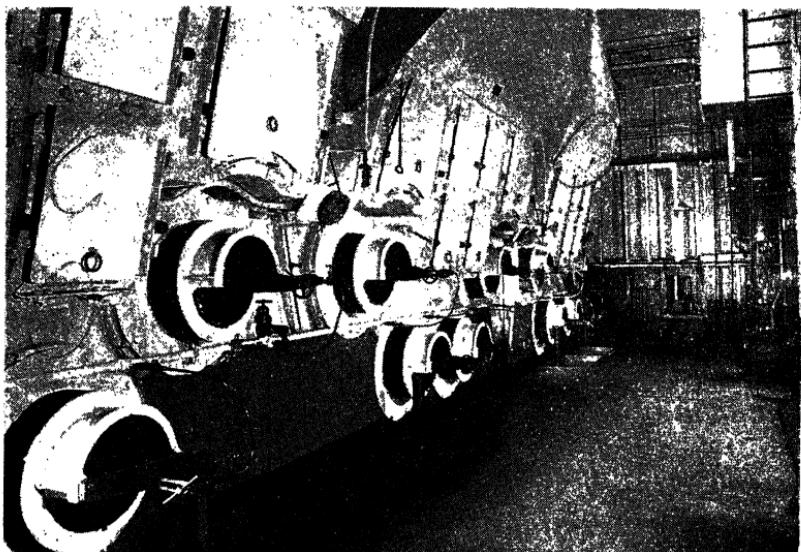


FIG. 85.—Boiler room of 6,000-ton freighter equipped with mechanical oil burners. (*Courtesy of Bethlehem Steel Corporation.*)

which has a range of temperature application up to 800°F.; a readily volatilized liquid, such as alcohol, which can be utilized

up to a maximum of 600°F.; and a nonvolatile liquid, mercury being almost universally used, with an application range extending up to 1200°F.

Capillary tubing, used to connect bulb with body, is of small diameter, so that only a small amount of the fluid is contained within it. Standard instruments are regularly equipped with 5 ft. of tubing, although special provision can be made for longer tubing, up to a maximum of 200 ft. for gas or vapor, or 50 ft. for mercury. The capillary tubing is enclosed in a protective bronze-armored tubing having flexible qualities.

The body consists essentially of a Bourdon tube, which translates pressure variations in the bulb into movement of a pointer

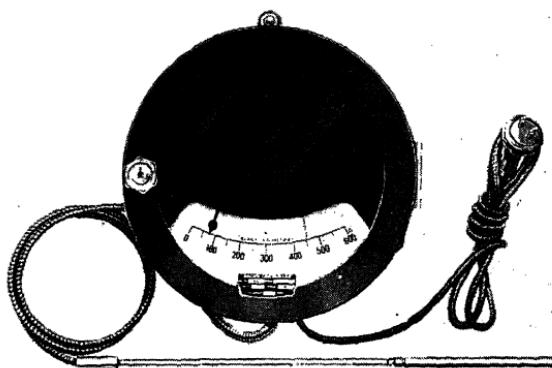


FIG. 86.—Brown indicating control thermometer.

or pen to either indicate or record temperatures. In the indicating type, the pointer merely moves over a temperature scale, while, in the recording type, a chart is moved by clockwork so that the pen may draw a line of temperature upon it. When the instrument is to be a controller as well as a thermometer, then the body contains also the mechanism which actuates electrical switching of the oil-burner parts in accordance with indicated temperature in the furnace. The time chart and controlling mechanism are driven by means of an electric clock movement (synchronous motor), which, of course, implies the availability of alternating current. When the plant power is direct current, a small rotary converter must be used to supply sufficient alternating current for the control system.

The desired temperature to be maintained in the furnace is indicated on the control instrument, by the control index which is manually set to the correct temperature position by means of a suitable knob. The electric clock periodically causes a "feeler" mechanism to determine the relative positions of the temperature pen or pointer and the control index, and should they be at

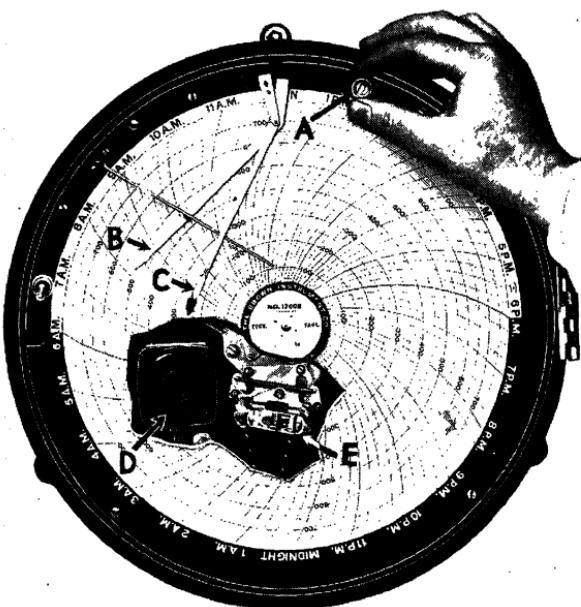


FIG. 87.—Brown recording control thermometer. *A*, knob for setting position of control index *B* which can be set at any temperature in the range of the instrument; *C*, temperature recording pen; *D*, synchronous motor; *E*, mercury-tube burner switch.

variance, electrical contacts are made which will cause the oil-burner firing rate to be adjusted accordingly. This control mechanism is explained in more detail on page 177 under the discussion of pyrometers.

Of the three types of thermometers, the mercury filled is the most desirable, because the expansion and contraction occurring in this liquid create more power than do either the gas or vapor, so that more delicate changes can be detected and recorded. It is therefore preferable at all temperature ranges where other instruments can be used at all, and must be used in the range

between 800 and 1200°F. There are two limitations to the use of thermometric controllers: limitations on distance between the instrument and the furnace, and limitations on temperature range. Whenever either of these two limiting values must be exceeded, then an application must be made of a pyrometric controller.

The thermoelectric pyrometer depends for its functioning upon a well-established fact of physics that when two rods of dissimilar metals are welded together at one end, and the welded junction is heated, an electric current is set up, and an electric potential difference created between the free ends of the rods. Since this induced electromotive force (e.m.f.) is proportional to the temperature of the junction, if suitable apparatus be provided for measuring the e.m.f., an indication of the temperature at the junction will be had.

The two metallic rods of dissimilar metals, with the welded junction, are called a thermocouple. Various combinations of metal have been evolved for various ranges of temperatures to be measured. The three most commonly used combinations, and their temperature ranges are:

Symbol	Positive wire	Negative wire	Max. temp.
IC	Iron	Constantan (60% Cu, 40% Ni)	1400
MA	Nickel chromium (90% Ni, 10% Cr)	Nickel aluminum (94% Ni, 2% Al, 3% Mn, 1% Si)	2000
QR	Platinum rhodium (90% Pt, 10% Rh)	Platinum	3000

The component parts of a millivoltmeter-type pyrometer are: a thermocouple, extension lead wires and a millivoltmeter calibrated to read directly in degrees temperature. There are no external sources of electricity, such as batteries, used in the circuit; the e.m.f. induced at the hot junction of the thermocouple is the only electrical current involved. Because the e.m.f. resulting from the hot junction is very small, an extremely sensitive galvanometer is used for the millivoltmeter. For accurate results, the terminals of the extension wires which connect to the

instrument, and the instrument itself, should be at normal room temperature, about 75°F.

In order to keep the instrument away from the heat of the furnace, it is frequently located at some distance from the furnace; the distance may range from a few feet up to many hundreds of feet, or even a thousand or more. Since a thermocouple generates a fixed e.m.f., it follows that the current passing through the millivoltmeter will be affected by the length of the wires. It is therefore necessary that an instrument be calibrated for a

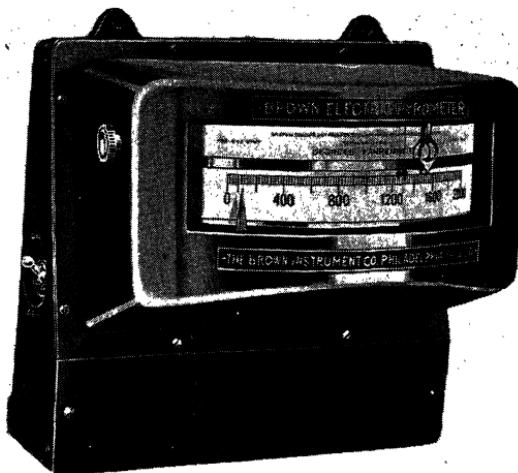


FIG. 88.—Brown indicating control millivoltmeter pyrometer.

specific length of extension wires. It is also possible, by means of balancing resistances, to use a single instrument to read temperatures of several couples, although only one thermocouple may be connected at any one time. The balancing of the circuits by resistances permits of the several furnaces being at varying distances from the instrument.

Pyrometers may be indicating only, or indicating and recording. Either type in turn may also be controlling. The controlling instruments are equipped with suitable devices for setting a pointer to the required furnace temperature on the instrument's scale. Another part, called the finder, is automatically moved up every 15 sec. to locate the temperature needle with respect to the pointer. Should the actual furnace temperature be too

low, the finding mechanism strikes the needle, which action disengages a clutch, so that the finder will not rise farther. At the same time an electric switch is closed, in the oil valve or burner circuit, so that more heat is put into the furnace. Should the finder, upon rising, strike the needle in the same position as the pointer, then no action occurs. Should the finder rise farther than the neutral position, then the temperature is too high, and again an electric contact is made in order to bring about a sequence of operations to reduce the firing rate.

In the potentiometer type of pyrometer, a similar thermocouple is used, but the instrument does not measure the induced e.m.f. directly. Instead, a counter electrical current is introduced, from a dry cell, and this opposing current is automatically adjusted to just oppose the intensity of the induced e.m.f. The slide wire which controls the opposing e.m.f. is rotated in order to bring into circuit the required amount of resistance. The rotation of this slide wire gives the temperature reading. The controlling feature is about the same as in the millivoltmeter type.

The controlling mechanism can also be of the modulating style instead of the contact closing or opening style. Here, the variations of furnace temperature are translated into motion by a reversing-type motor controller, which in turn will regulate the amount of opening of oil, air or steam valves, as the case may be. This modulating action, however, is available only in pyrometers; it cannot be obtained in controlling thermometers discussed earlier in this chapter.

Thermocouples themselves are manufactured in a variety of shapes and styles, for the diverse uses which are encountered. In all cases the couple must be protected from the chemical and physical effects of the furnace atmosphere. This is accomplished by various types of housings, coverings or sheaths. The connection terminals must similarly be protected by being enclosed in terminal boxes into which the extension lead wires may come for connection to the ends of the couple.

For temperature ranges above 3000°, pyrometers must be of the optical or radiation type. These depend upon either the light or total heat radiated from hot bodies.

Thermometers, in general, with standard lengths of capillary tubing, cost only one-third to one-half as much as pyrometers. On the other hand, they are restricted in their installation loca-

tions, and a single instrument can be used for only one furnace, in permanent installations. Pyrometers are more accurate, have closer control differentials and give better flexibility in installa-

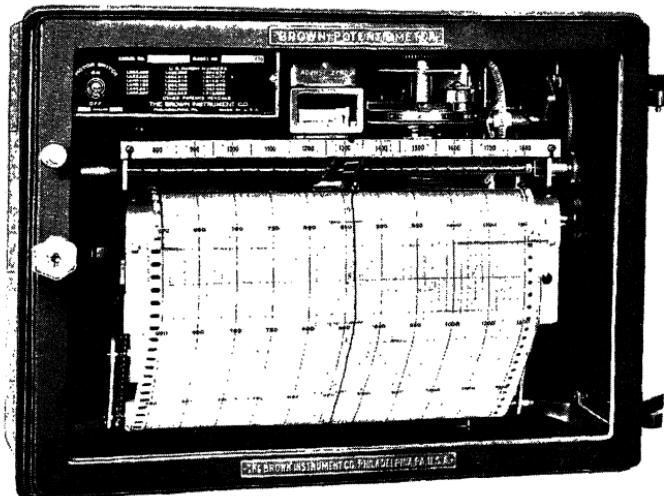


FIG. 89.—Brown single-point recording control potentiometer.

tion. A single instrument can be utilized for controlling up to three furnaces or for recording temperatures in as many as six furnaces.

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CHAPTER IX

FANS AND ELECTRIC MOTORS

FANS FOR OIL BURNERS

Many oil burners of all groups, domestic, commercial and industrial, supply at least part of the total air of combustion as primary air, and generally this primary air participates also in atomizing the fuel oil. This primary air must be supplied by mechanical means, either fan, blower or compressor. In many burners the fan or other air-blown mechanism is an integral piece of the assembly, and a great deal of care and thought is given to its design and construction. Because of the importance of air supply to combustion of fuel oil, it is worth while to consider the general engineering theory of fans and compressors, although in this book the discussion of this subject must necessarily be very limited. One authority on the subject gives no less than 14 factors that affect fan design and performance and that must be considered before a satisfactory answer may be expected as the solution of a fan problem. Others state that no theoretical solution of a fan problem can be worked out in advance, but that only trial-and-error development of a specific fan for a specific job will yield the right fan in any one case. Between these two extremes the oil-burner designer can steer a medium course by observing the fundamental engineering principles and modifying these by the known facts of experience and observation.

In general, air-blowing devices can be classified as fans and as compressors, although the term blower is applied rather loosely to both. While there are really two kinds of fan, centrifugal and disk or propeller, only the former, or centrifugal, is encountered in oil-burner work. In compressors, the positive-rotary type is most frequently used; reciprocating or piston compressors and centrifugal or turbo compressors have little utility in fuel-oil equipment. Centrifugal fans of conventional design develop air pressures ranging up to 10 in. water gauge, while in special designs, particularly for the horizontal-rotary burner, pressures

go as high as 25 in. water gauge. Rotary compressors can pump air to pressures reaching up to 15 lb. per sq. in. In the rare instance where high-pressure atomizing nozzles are used, it becomes necessary to employ reciprocating compressors or multi-stage turbo compressors to build up pressures attaining values between 75 and 175 lb. per sq. in.

Present practice in the manufacture of conventional centrifugal fans seems to have standardized on the multiblade fan, in three styles of blade: forward curved, straight or radial, and backward curved. Multiblade fans are characterized by wide shallow blades—the length being only a very short portion of the fan radius. The forward-curved blade is most commonly used, for the reason that the air leaving this fan actually has a higher velocity than the tip speed of the wheel itself. This feature makes possible low rotative speeds of the wheel, which in turn means quietness of operation. Hence the forward-curved multiblade fan is widely used in domestic gun-type burners, where relatively large quantities of air are desired at low pressure. Multiblade fans with radial tips are rarely used for any purpose, and backward-curved blades find their chief value in cases where avoidance of motor overload is imperative. But since motor overload seldom occurs on an oil burner from fan performance, and since the characteristics of a forward-curved blade are much more valuable to an oil burner than mere avoidance of motor overload, the forward-curved blade is used almost exclusively in multiblade fans in oil-burner practice.

Figure 90 illustrates a typical forward-curved multiblade fan. The pressure developed by this fan originates in both the centrifugal action of rotation and the velocity acquired from the blades themselves. This latter is partially converted into pressure head by the scroll or involute casing. The involute gives an increasing cross-sectional area of discharge chamber as it proceeds around the fan, and its shape must be carefully determined to minimize losses which always occur when fluids are moved through conduits of varying direction and area. The same factors must be considered in forming the entrance into the wheel, for unless smooth or streamlined shapes are used, high losses will occur to reduce the fan's efficiency and capacity. Fan manufacturers have conducted extensive experiments on these phases of fan design, and their recommendation should always be used when the casing

itself is part of the burner assembly and is to receive a standard wheel made by a fan manufacturer. Abrupt changes in air direction or travel should be avoided, as well as sudden changes of cross section.

The multiblade fan with forward-curved blades is used exclusively in domestic gun-type burner, whether high- or low-pressure atomizing, and is also used extensively in vertical-rotary wall-flame vaporizing burners. Since these burners furnish almost all the combustion air as primary air, the required capacity of

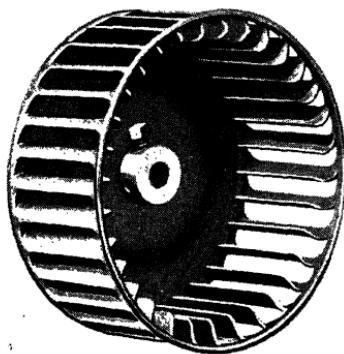


FIG. 90.—Multiblade forward-curved fan wheel. (*Courtesy of American Blower Company.*)

the fan can be calculated very quickly from the maximum oil-burning capacity of the burner, in any case. Since 1500 cu. ft. of air are required for theoretical combustion of a gallon of oil composed of 85 per cent carbon and 15 per cent hydrogen, if provision be made for 50 per cent excess, then the required cubic feet will be

$$Q = \frac{\text{g.p.h.} \times 1500 \times 1.5}{60}$$

in which Q is in cubic feet per minute and g.p.h. is gallons of oil per hour.

In selecting a fan to deliver any required capacity of air, reference is usually made to manufacturers' catalogue tables. But selection from such tables can only be made with accuracy if the resistance which the fan must work against is known. It is customary to express this resistance in inches water gauge. The resistance is the friction through the burner air tube plus the

losses of head due to eddy currents, changes in cross section of air conduits, etc. A more thorough discussion of flow of fluids is given in a later chapter on Fuel Oil Pumps and Pumping; much of the hydraulic theory there given is applicable to air flow, but the actual working data on gas flow have not been worked out as completely or accurately as for liquids. Hence it is necessary to work out many air-flow problems by assuming certain conditions to exist, selecting trial equipment accordingly, and then testing to see what results are obtained. From these data any necessary modifications in design can be made and a better choice of equipment then made.

To illustrate the effect of conditions on fan performance, the following data are given from catalogue material of a well-known fan manufacturer. The figures cover the air delivery of a standard forward-curved multiblade centrifugal wheel, 6-in. diameter by 3 in. wide, with 32 blades, showing air deliveries in cubic feet per minute against static discharge pressures (S.P.) in inches water gauge, at two different speeds:

R.p.m.	0.2 in. S.P.	0.4 in. S.P.	0.6 in. S.P.	0.8 in. S.P.	1.0 in. S.P.	1.4 in. S.P.	1.8 in. S.P.	2.2 in. S.P.	2.6 in. S.P.
1750	318	266	206						
3400	690	667	645	617	592	540	470	405	330

In standard burner design, the air tube seldom exceeds 4 in. in diameter, and the maximum capacity of a domestic burner should logically be 6 g.p.h. The required fan capacity would in this case be 225 c.f.m., so that the maximum allowable pressure loss would be, from the above table, about 0.5 in. water. With so small an allowable loss, it is essential that the fan inlet be streamlined, that the involute casing be properly designed and that the air tube be fairly short. Needless to say, the discharge from the burner air tube must be free, but since most fireboxes will be below atmospheric pressure, there is seldom difficulty on this score.

Centrifugal fans for horizontal-rotary burners are a special case of this general type of wheel. Here the need is for relatively small quantities of air at high pressure. Since the horizontal rotary uses as much as 90 per cent secondary air (usually induced

by natural draft), the fan must deliver only 10 or 15 per cent of the total required combustion air. But the pressure of delivery must be high enough to keep in suspension the oil spray being whirled off the atomizer periphery.

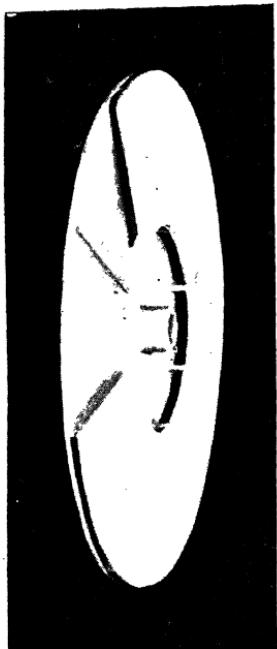


FIG. 91.—Centrifugal fan for horizontal-rotary oil burner. (Courtesy of Ace Engineering Company.)

In the horizontal-rotary fan, high pressure is secured by large diameters and high rotative speeds (3400 r.p.m. being customary). Since the required volume is small, the blades can be deep, and so are brought to within a few inches of the shaft, the fan inlet thus being much smaller than in the case of the multiblade. Few blades are used, and the fan is narrow (Fig. 91). All this tends to give the fan rigidity and strength, so that lightness can be secured with thin fan plates without danger of the fan rupturing. Again, since efficiency is not a primary consideration when blowing small air quantities, the blades are straight, not curved, thus making manufacturing simpler and less costly.

The following table gives typical data, from tests on a series of centrifugal fans for horizontal-rotary burners, showing the effect on static no delivery (S.N.D.) of increasing fan diameter while holding constant the speed (3400 r.p.m.) and number of blades (8).

TABLE XX.—DATA ON CENTRIFUGAL FANS

Fan Diameter, In.	Pressure S.N.D., In. Water
8.5	3.0
12	7.0
14	8.75
15.5	10.25
17	13.0
18	15.0

With two 18-in. fans in series, *i.e.*, for a two-stage fan, the devel-

oped S.N.D. was 28 in. water, so that for a two-stage fan the total pressure was 1.87 times that of a single-stage fan.

Positive-rotary compressors are similar in design and construction to rotary pumps, with the necessary modifications for handling gases instead of liquids.

In single stages, pressures as high as 15 lb. per sq. in. can be secured. They are used in oil-burner work to supply air for low-pressure air atomizing nozzles, and as a rule are not called upon to develop pressures exceeding 5 lb. per sq. in. This type of equipment works at peak efficiency when the pressures are between 3 and 5 lb., so that for oil burners

they perform at their best. The horsepower required can be roughly computed as 1 hp. per 200 cu. ft. free air delivered per minute at a pressure of 1 lb. per sq. in., although, of course, this will vary with other factors which affect performance. But for a

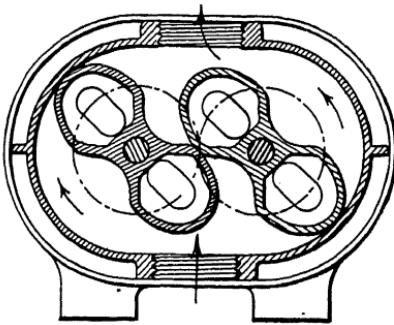


FIG. 92.—Rotary positive blower.
(Courtesy of Roots-Connorsville Blower

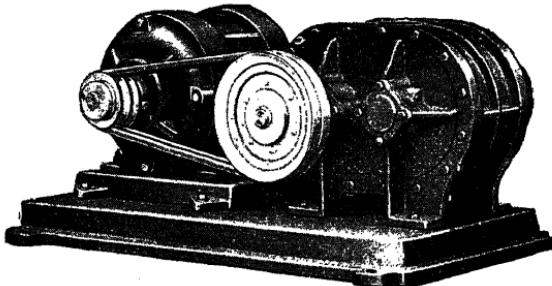


FIG. 93.—Motor-driven rotary blower set. (Courtesy of Roots-Connorsville Blower Corporation.)

quick computation, it may be assumed that power varies directly with volume and pressure on the above ratio. Rotary compressors are made in many styles, with capacities ranging from 10 up to 10,000 c.f.m. They are generally simple and rugged, giving long service with little maintenance expense. The rotary compressor has the distinct advantage over a centrifugal fan that it will handle a constant volume under varying conditions of pressure and discharge.

Centrifugal or turbo compressors are built the same as centrifugal pumps, and in some respects resemble the centrifugal fan above described for horizontal-rotary burners. In fact, the latter and the multiblade fan are but special cases of the centrifugal turbo compressor. In strict industrial applications, the turbo compressor is made in large sizes and in multiple stages, so that it will develop pressures as high as 175 lb. per sq. in. Reciprocating or piston compressors also can be used to secure high compression, up to equivalent values, but these two machines are

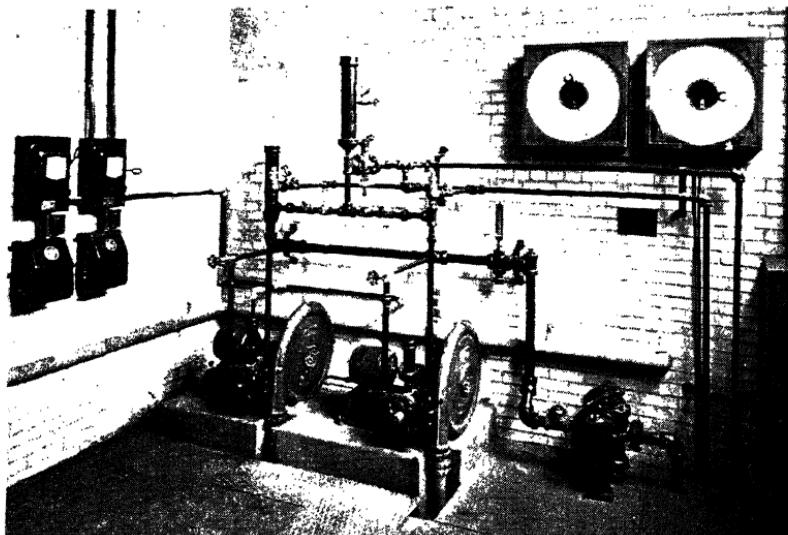


FIG. 94.—Centrifugal blowers. (*Courtesy of Simplex Oil Heating Corporation.*)

seldom required except where high-pressure atomizing nozzles are used. In such cases, steam is the common atomizing medium, but at those times when steam is not available, as when starting up a boiler plant, then high-pressure air must be used till steam is raised. Under such conditions of high compression, the water vapor contained in air will condense out, so that provision must be made for its collection and removal from the air-pumping system.

Three pressures must be considered in a column of moving air: static, velocity and impact. The first pressure results from the compression imparted to air as it passes through a fan or compressor and can be measured by a pressure gauge or manometer connected into the duct wall. The velocity pressure represents

the energy of movement and can be translated into corresponding pressure by stating the height of a column of air or water which will impart an equivalent head. The impact pressure is the sum of both static and velocity heads.

TABLE XXI.—SPECIFIC WEIGHT OF AIR AT VARIOUS TEMPERATURES,
PRESSURES AND DEGREES OF HUMIDITY*

Temp., °F.	Weight 1 cu. ft. dry air (at 14 lb. per sq. in. or 28.5 in. Hg pressure), lb.	Increase or decrease for each 0.1 lb. change in pres., lb.	Increase or decrease for each 1 in. Hg change in pres., lb.	Decrease for each 10% in- crease in rel- ative humidity, lb.
32	0.07688	0.000549	0.002698	0.000019
35	0.07642	0.000546	0.002681	0.000021
40	0.07565	0.000540	0.002654	0.000025
45	0.07490	0.000535	0.002628	0.000030
50	0.07417	0.000530	0.002602	0.000035
55	0.07340	0.000525	0.002580	0.000040
60	0.07272	0.000520	0.002554	0.000051
65	0.07203	0.000515	0.002530	0.000059
70	0.07134	0.000510	0.002506	0.000070
75	0.07068	0.000505	0.002482	0.000081
80	0.07003	0.000500	0.002457	0.000095
85	0.06938	0.000495	0.002432	0.000111
90	0.06875	0.000490	0.002408	0.000127
95	0.06811	0.000485	0.002384	0.000147
100	0.06752	0.000480	0.002359	0.000172
105	0.06694	0.000475	0.002334	0.000199

* From Marks' "Mechanical Engineers' Handbook," McGraw-Hill Book Company, Inc.

Pressures in air streams are most conveniently measured in terms of inches of water since the water gauge or manometer is the most practical instrument for making determinations. One inch of water equals 0.577 oz. per sq. in., or 1 oz. per sq. in. equals 1.732 in. water. For air at 70°F. and 29.92 in. barometric pressure (weight per cubic foot 0.0749 lb.), the height in feet of a column of air corresponding to any pressure in inches of water is given by $H = 62.5h$, H being feet of air and h being inches of water.

OIL BURNERS

TABLE XXII.—WEIGHT OF SATURATED AIR IN POUNDS PER CUBIC FOOT AT
DIFFERENT BAROMETRIC PRESSURES*

Temp., °F.	Barometric readings, in. Hg								
	28.5	29.0	29.5	29.7	29.9	30.1	30.3	30.5	31.0
30	0.07703	0.07839	0.07974	0.08028	0.08083	0.08137	0.08191	0.08245	0.08381
35	0.07621	0.07756	0.07890	0.07943	0.07997	0.08051	0.08104	0.08158	0.08292
40	0.07541	0.07674		0.07859	0.07913	0.07966	0.08019	0.08072	0.08205
45	0.07461	0.07592	0.07724	0.07776	0.07829	0.07881	0.07934	0.07986	0.08118
50	0.07381	0.07512	0.07642	0.07694	0.07748	0.07798	0.07850	0.07902	0.08032
55	0.07302	0.07431	0.07560	0.07612	0.07663	0.07715	0.07766	0.07818	0.07947
60	0.07224	0.07352	0.07479	0.07530	0.07581	0.0763	0.07683	0.07734	0.07862
65	0.07145	0.07272	0.07398	0.07449	0.07499	0.07550	0.07600	0.07651	0.07777
70	0.07067	0.07192	0.07317	0.07367	0.07417	0.07467	0.07518	0.07568	0.07693
75	0.06988	0.07112	0.07236	0.07286	0.07335	0.07385	0.07434	0.07484	0.07608
80	0.06909	0.07032	0.07155	0.07204	0.07253	0.07302	0.07351	0.07400	0.07523
85	0.06829	0.06950	0.07072	0.07121	0.07170	0.07218	0.07267	0.07316	0.07437
90	0.06748	0.06868	0.06989	0.07037	0.07085	0.07133	0.07182	0.07230	0.07351
95	0.06665	0.06785	0.06904	0.06952	0.07000	0.07048	0.07095	0.07143	0.07263
100	0.06581	0.06700	0.06818	0.06866	0.06913	0.06960	0.07008	0.07055	0.07174

* From Marks' "Mechanical Engineers' Handbook," McGraw-Hill Book Company, Inc.

TABLE XXIII.—SPECIFIC HEATS OF DRY AND SATURATED AIR*

Temp.,	Specific heat of	
	Dry air	Saturated
60	0.2417	0.244
65	0.2417	0.2447
70	0.2417	0.2452
75	0.2418	0.2458
80	0.2418	0.2466
85	0.2419	0.2474
90	0.2419	0.2486
95	0.2420	0.2498
100	0.2420	0.2512
105	0.2420	0.2526

* From Marks' "Mechanical Engineers' Handbook," McGraw-Hill Book Company, Inc.

Pressure head may be translated into velocity from the basic equation, $v = \sqrt{2gH}$, v being in feet per second. Using V as feet per minute velocity, $V = 4000\sqrt{h}$. It follows, therefore, that a pressure or head of 1 in. of water will produce a velocity of 4000 ft. per min.

Volume of delivery can be readily calculated if the velocity and cross section of the delivery duct or tube are known, the formula being c.f.m. = $Q = AV$, A being in sq. ft., and V velocity in feet per minute, as before. Where no data are at hand pertaining to an air stream, an anemometer will give the air flow.

Power required to move an air stream is a function of the total volume moved and the combined static and velocity pressures, or impact pressure. Air horsepower is expressed as a.h.p. = $5.2 \times Q \times h_i / 33,000$. Mechanical efficiency is the ratio of air horsepower to the actual or brake horsepower expended; hence brake horsepower can be calculated from b.h.p. = a.h.p./eff. If an air stream is discharging from or through an oil burner without much resistance to be overcome, as in a gun-type burner, then the air quantity can be translated back to velocity head by $h_s = [Q/(4000 \times A)]^2$. Adding to this about 0.5 in. water gauge for h_s , or head lost in overcoming friction and other losses, then h_i or impact pressure is known.

The following generalizations will be helpful in considering fan problems. For centrifugal fans of all types and sizes, the volume delivered varies directly with speed N (r.p.m.), the pressure developed varies as N^2 , and the air horsepower varies as N^3 , also the brake horsepower as N^3 . For any given peripheral speed and resistance pressure, the volume delivered varies as the square of the diameter D^2 . The required horsepower also varies as D^2 .

ELECTRIC MOTORS FOR OIL BURNERS

Electric motors for oil-burner applications are classified first into the two current groups, direct current and alternating current, usually designated respectively as d.c. and a.c. These two types of current are so radically different in their nature that motors made for operation on one are totally different from

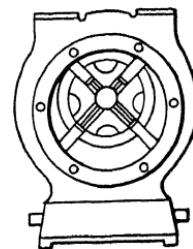


FIG. 95.—Rotary air compressor.

motors made for operation on the other. Direct current is a form of electrical energy in which the electrical pressure or potential is impressed over the distributing lines at a constantly maintained level and in a uniform direction, in which the positive line is conceived of as carrying the potential from the energy source to the driven device and the negative line as returning the potential from the driven device to the energy source. In the case of alternating current, however, the electrical potential is constantly varying in both intensity and direction, so that it varies from zero to a maximum while flowing in one direction, then diminishes to zero again, at which time the flow reverses in direction and increases to a maximum and then decreases to zero while flowing in the opposite direction. The time interval during which the current changes from zero to one maximum, back to zero, up to the other maximum, and again back to zero, is called a cycle. The number of cycles per second occurring with any energy supply is called the frequency. Commercial frequencies used in the United States are 25, 50 and 60 cycles per second, 60 cycles being the predominating frequency. Electrical potential or pressure is measured in volts, the customary potentials encountered being 110 or 115, 220 or 230 and 440 volts. Intensity of current flow is measured in amperes. The product of volts multiplied by amperes gives the apparent power, which is measured in watts. Hence a current of 5 amp. flowing at a potential of 110 volts would give an apparent power of 550 watts. Since a mechanical or electrical horsepower is 746 watts, the power here would also be 0.73 hp. Electric motors must be investigated and analyzed, in order to determine their suitability to oil-burner application, from the viewpoints of type of current, voltage, power, frequency, speed, torque, operating characteristics and mechanical construction.

DIRECT-CURRENT MOTORS

Direct current is no longer extensively used in this country, but it is still encountered sufficiently often so that manufacturers must be in a position to supply burners for direct-current operation. Direct-current motors are of three types: series, shunt and compound. All three types have two windings—a field winding and an armature winding—and in all three types current is supplied to the armature winding by means of a commutator which

is part of and rotates with the armature while stationary brushes press up against the commutator and make the electrical contact through which the current flows into and out of the armature windings. In the series motor, the two windings, field and armature, are connected electrically in series, so that all the current taken by the motors flows in sequence first through the field and then through the armature. In the shunt type, the motor current divides, part going through the field and part through the armature. In the compound type, the field has two

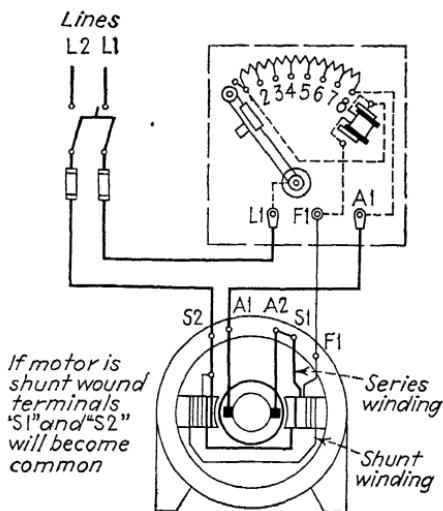


FIG. 96.—Connections for manual starting rheostat on compound direct-current motor.

windings, one of which takes separate current as in the shunt motor while the other takes the armature current also, as in the series motor.

Series motors have a high starting torque, *i.e.*, they can start up while heavily loaded, but the speed varies inversely as the load current, so that at light loads the motor speeds up and at heavy loads the motor slows down. Because of this variable speed, the series motor is seldom used on oil-burner work, where the usual requirement is for a constant speed motor.

Shunt motors are the most common type of direct-current motor for general use, and are widely used in oil-burner application, especially in large sizes (over 1 hp.). They maintain almost

constant speed regardless of load, the variation from no load to full load being only 5 per cent.

Compound motors have characteristics resembling both series and shunt, the starting torque being good, and the speed regulation being more stable than for the series and not quite so uniform as with the shunt. It is frequently used for heavy duty work of short duration, as on a punch press or air compressor. The

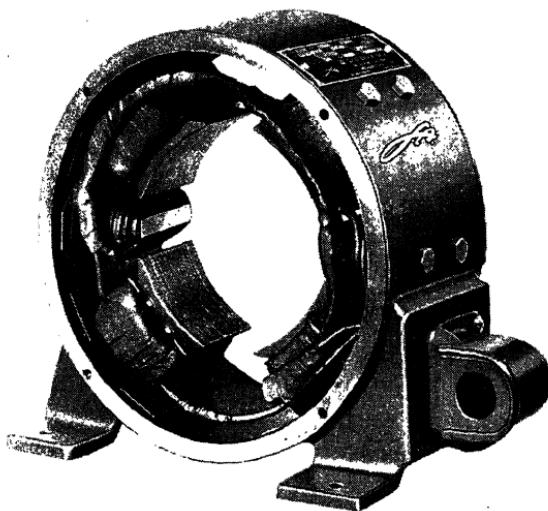


FIG. 97.—Field windings and frame for direct-current motor. (*Courtesy of Louis Allis Company.*)

compound motor is used for oil-burner work when the motor size is fractional (under 1 hp.).

Small sizes of direct-current motors, up to possibly $\frac{1}{2}$ hp., may be started by being placed directly across the line if the speed does not exceed 1750 r.p.m. For larger sizes and for speeds above 1750 in the $\frac{1}{2}$ hp., direct-current motors must be brought up to speed gradually by means of starting rheostats which are connected to control the armature current. For manual starting a typical starting rheostat with diagram of connections is illustrated in Fig. 96. For automatic starting, it is necessary to use a dashpot-type magnetic starter, as described in Chap. X.

Commutation in direct-current motors is one of the principal mechanical problems. A commutator is built up of alternate

segments of mica insulators and copper. Motor manufacturers attempt to use mica having the same wearing qualities as the copper so that both will wear away at an equal rate. Sometimes the copper wears down more rapidly, however, with the result that the commutator surface on which the brushes press becomes uneven, with the mica protruding beyond the copper. This, of course, causes extremely rapid wear on the brushes which under these conditions will require frequent replacement. When a commutator reaches this condition the armature must be placed in a lathe and the commutator cut down to an even surface

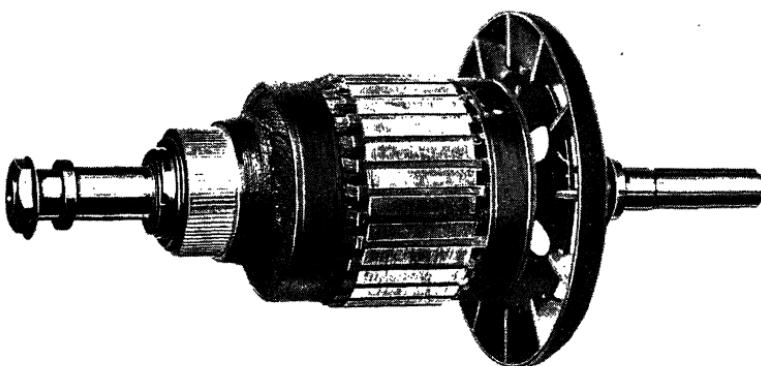


FIG. 98.—Armature, commutator, shaft and end bell for direct-current motor.
(Courtesy of Louis Allis Company.)

again. When the tendency for the copper to wear faster than the mica becomes marked, then the mica is undercut, so that the copper protrudes beyond the mica. This will permit of a period of operation during which the mica will not come into contact with the brushes. In general, commutation difficulty increases with the speed. Some manufacturers of horizontal-rotary burners, rather than face the problem of commutation on direct-current motors operating at 3450 r.p.m., will use 1750-r.p.m. motors and belt or gear these motors to the burner shafts with a 2:1 speed ratio, so as to drive the shaft at 3450 r.p.m.

The bearings on direct-current as well as on alternating-current motors should be sleeve bearings whenever possible, as ball bearings are not nearly so quiet in operation. Phosphor-bronze bearings are preferable to babbitt or other soft-metal bearing linings as the latter wear too quickly.

For pressure-atomizing and other domestic-type oil burners a very common size of motor is $\frac{1}{6}$ hp., running at 1750 r.p.m. When the burner is to operate under automatic control, it is customary to effect ignition from a high-tension spark. To supply a source of alternating current for the ignition transformer, many motor manufacturers equip this size of direct-current motor with special connections and set of collector rings and brushes to generate sufficient alternating current for ignition. This generated current is usually about 70 volts 30 cycle, and a $\frac{1}{6}$ -hp. motor can deliver about $3\frac{1}{2}$ amp. of this current. Transformers can be had to meet these current specifications for electric ignition purposes.

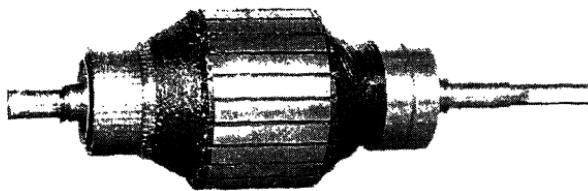


FIG. 99.—Armature of small direct-current motor showing collector rings for alternating current generated to operate ignition transformer. (*Courtesy of Century Electric Company.*)

The direction of rotation of all direct-current motors can be reversed by interchanging the line connections to *either* the field or the armature, but not to *both*. On most direct-current motors, the simplest method of reversing the rotation is to interchange the two lead wires to the brush holders. When direct-current motors are used in direct drive, *i.e.*, where the load is carried by extension of the motor shaft, it is sometimes the practice to screw the motor shaft into the load shaft, to facilitate removal of the armature should rewinding become necessary. In applications such as this, it is imperative that the direction of rotation, when the burner is being placed into service for the first time after installation or after any electric work is performed on the motor or line, be determined before the motor is allowed to attain speed. This can of course be done by just making momentary contact on the line to permit the motor to give just a slight impulse of turning. If wrong, the rotation should be reversed immediately. Otherwise, should the motor attain speed in the wrong direction,

the screwed connection on the shafts can be loosened, and the armature will promptly unscrew from the burner shaft and cause considerable damage.

Many oil-burner designs, especially the horizontal rotary, are such that the burner fan pulls air through the motor and so cools it. Otherwise the armature shaft should be equipped with a fan to effect ventilation and so prevent overheating of the motor. All

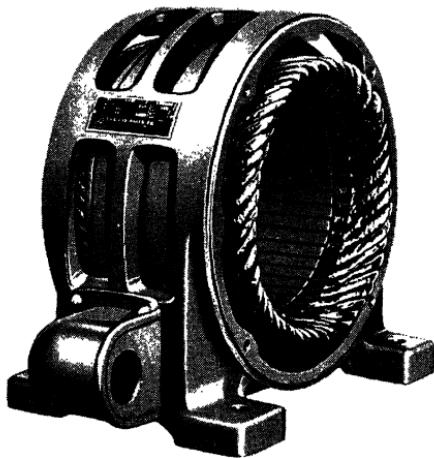


FIG. 100.—Stator windings and frame for standard squirrel-cage induction motor, polyphase. (*Courtesy of Louis Allis Company.*)

oil-burner motors should be of the so-called long-hour-duty type, which means over 1000 hr. service per year.

ALTERNATING-CURRENT MOTORS

Alternating-current motors fall into two general types: polyphase and single phase. Of the polyphase, three phase is by far the most common; two-phase current supplies are quite rare. There are two kinds of polyphase motors, known as the synchronous and induction. Synchronous motors are used only in very large sizes for major industrial uses; by far the most common kind of motor used in present-day practice is the induction motor, which again has two classifications: squirrel cage and wound rotor. Only the squirrel-cage induction motor is encountered in ordinary oil-burner work. The single-phase motor has three

classifications, known as the repulsion-induction, split phase and capacitor. All three of the single-phase motors are used for oil burners. The direct-current series motor can also be used on alternating current, and has many of the properties of the direct-current motor. The series motor, then called a universal, has application on alternating current only when very high speeds are required.

The squirrel-cage motor is without exception the most suitable and practical style of electric motor for ordinary applications. It has only one field winding, it has no commutator, it has no brushes, it has no starting winding on the armature, hence no moving contacts of any sort and is essentially a constant speed motor. Its name is derived from the resemblance which the

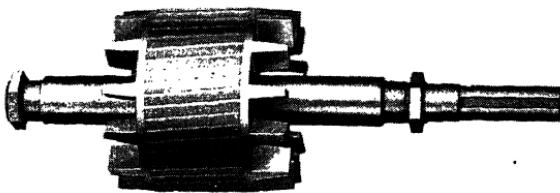


FIG. 101.—Rotor and shaft for polyphase induction motor, squirrel-cage. (*Courtesy of Louis Allis Company.*)

armature or rotor bears to a cage in appearance, for the rotor winding is built up of bars fastened into end rings. The entire motor is rugged and dependable in performance. Should the stator winding burn out it can be replaced or rewired at very moderate cost. Rotation is reversed by merely interchanging any two of the three lead wires (on 2 or 3 phase). Its starting current is low, and the motor can always be started by placing it directly across the line (up to at least 5 hp.). While its power factor is low, the item of electric energy consumption is of small importance compared to operating mechanical features. At that, the squirrel-cage motor can develop efficiencies of almost 90 per cent, from 50 to 125 per cent load. It is available in sizes as small as $\frac{1}{4}$ hp. or even smaller, and should certainly be selected for all applications where polyphase power is available.

In the case of the single-phase motor the story is not nearly so satisfactory. The very fact that at least three types of single-phase motors have been made available, some with subclassifica-

tions, indicates that there exists at least a variation of opinion on the matter, and as a matter of fact, there exists also a mechanical and electrical deficiency in the ability of single-phase motors to meet all requirements.

The most popular single-phase motor is the repulsion-induction. As is true of most single-phase motors, this motor runs as an induction motor, just the same as a polyphase. But it must be started in an electrically different manner. For starting, a second winding is placed upon the rotor, and while there is no

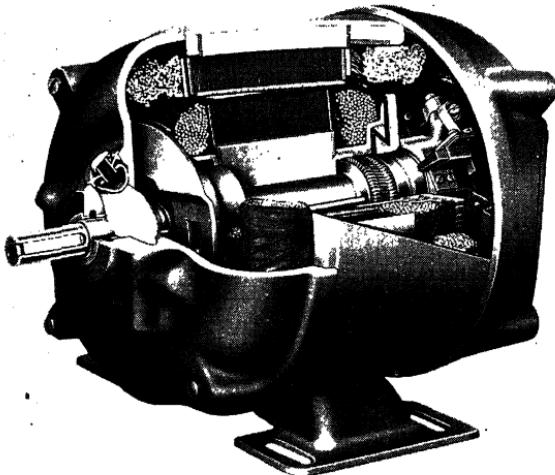


FIG. 102.—Sectional view of repulsion-induction motor, single-phase. (*Courtesy of Century Electric Company.*)

electrical connection between the rotor and stator, a commutator and set of brushes are necessary on this motor to permit of inducing a current in the rotor which permits the motor to start. As soon as the motor attains speed a short-circuiting device comes into contact with the commutator. It is not necessary here to discuss the electric theory of the repulsion-induction motor, or of any other motor, but it is self-evident that in order to short-circuit the commutator it is necessary to provide additional mechanical equipment over that needed on the squirrel cage. In addition, there is wear on the brushes, and hence replacement is needed from time to time. In spite of all this, the repulsion-induction motor is usually satisfactory for speeds up to 1750 r.p.m. At 3450 r.p.m., however, the brush and related problems become aggravated.

Split-phase motors would probably be the next best type of single-phase motor were it not for an operating characteristic which causes these motors to draw a very high starting current. These motors have no additional winding on the rotor; in fact the rotor is identical with and is interchangeable with the squirrel-cage rotor. There is, however, an additional winding in the stator, and this winding is very light and can carry current for only a few seconds. Its circuit is opened by a centrifugal mechanism as soon as the motor attains speed. While this centrifugal

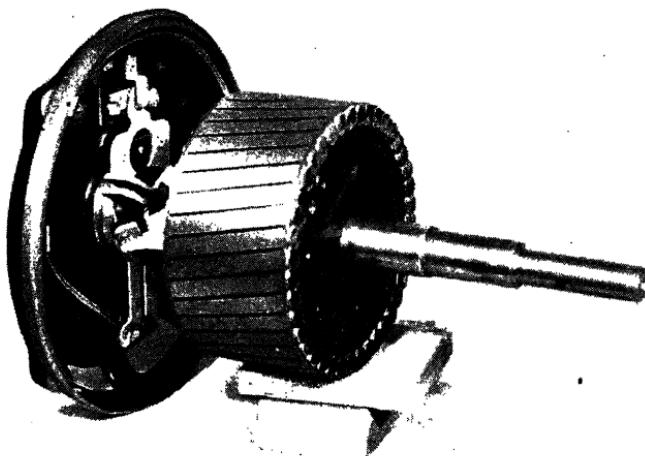


FIG. 103.—Rotor, shaft, and end bell of a split-phase motor, showing centrifugally operated switch for opening starting winding. (*Courtesy of Wagner Electric Corporation.*)

device does give some maintenance difficulty, it is much simpler than the starting mechanism of the repulsion-induction. But in starting, the split-phase motor draws many times the running current, and most public-service companies object to its use because the high starting current requires special provision of supply capacity and where this special provision is lacking (which is the usual condition), split-phase motors disturb the entire distributing system being served from a transformer on the public-service lines. Attempts have been made to install resistance coils in the starting winding circuit, but without marked success. In the $\frac{1}{6}$ -hp. size this motor is quite satisfactory as a substitute for the repulsion-induction, but above that its use is not advisable.

The most recent development of the single-phase motor is the capacitor type. While not new in theory, motor manufacturers made no attempt to exploit this class of motor until the repulsion-induction and split-phase motors showed their shortcomings, and one of the principal applications where the need for another motor was manifest was the oil burner. The capacitor motor has two subclassifications, capacitor-start-induction-run, and capacitor-start-capacitor-run. As the names might imply, these motors

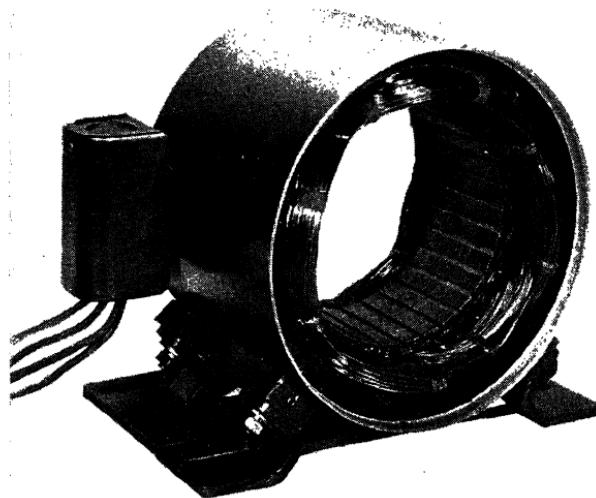


FIG. 104.—Stator and frame of a split-phase motor. (*Courtesy of Wagner Electric Corporation.*)

have an additional winding called the capacitor, and in the former case the capacitor winding is used only to start the motor which then runs as in induction motor, while in the second case, the capacitor winding remains in circuit as long as the motor is running. The capacitor-start-induction-run seems to be the more satisfactory motor for oil-burner work. In this motor the complicated brush and short-circuiting mechanism of the repulsion-induction motor is eliminated, and the high starting current objection of the split-phase motor is not present. While some development work remains to be done on the capacitor motor, it is now being offered in styles that are satisfactory for operation up to 3450 r.p.m. Sometimes the capacitor winding is incorporated in the stator housing on the motor itself and is opened by

centrifugal action, and sometimes it is separately housed for wall mounting, the winding circuit being opened from a switch in the separate housing which is actuated at full speed by induced current.

As to starting torque, the repulsion-induction motor and capacitor-start-induction-run motor both have high starting torques and both can be used interchangeably on the same

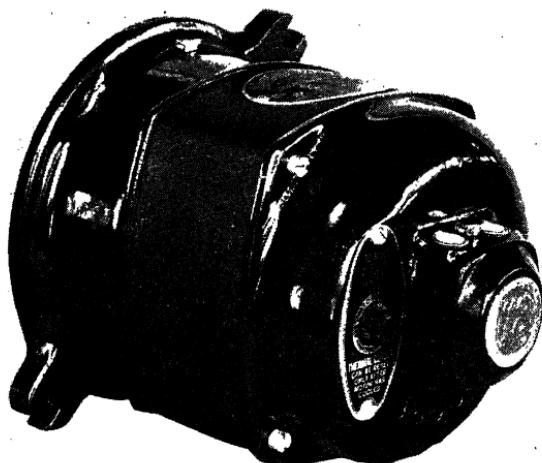


FIG. 105.—Flange-mounted motor for domestic gun type burner, with overload cutout and reset. (*Courtesy of Harvey-Whipple Inc.*)

application, insofar as torque is concerned. Both the split phase and capacitor-start-capacitor-run motors have low starting torques. However, on most burners, the principal loads are the fan and oil pump, and since neither fan nor pump offers much load on starting, torque is not an important starting characteristic for a motor for this use. Where the oil burner employs an air compressor, however, this factor must be considered, for compressors offer a high starting load.

The speed of all alternating current motors is determined by the frequency and number of poles in the motor, the formula being

$$\text{R.p.m.} = 120f$$

in which f is the cycles per second and p the number of poles.

Hence the possible speeds are limited by various combinations of frequency and poles, as shown in the table.

TABLE XXIV.—RELATION BETWEEN NUMBER OF POLES AND SPEED OF ALTERNATING-CURRENT MOTORS

Poles	R.p.m.		
	60 cycles	50 cycles	25 cycles
2	3600	3000	1500
4	1800	1500	750
6	1200	1000	500
8	900	750	375
10	720	600	300

The speeds shown in above table are so-called synchronous speed, and are really the speeds of the rotating fields induced in

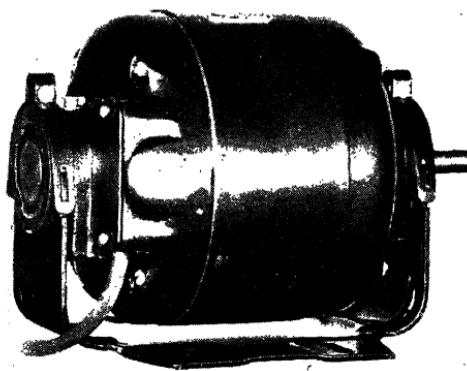


FIG. 106.—Split-phase motor with rubber-cushioned base. (*Courtesy of General Electric Company.*)

the motor windings by the impressed current. The rotor speed is less than the synchronous speed, the difference being known as the slip. Slip is usually expressed as the percentage which rotor speed bears to synchronous speed. At no load the rotor speed approaches synchronous speed, but as load is applied rotor speed reduces somewhat. Most induction motors will carry

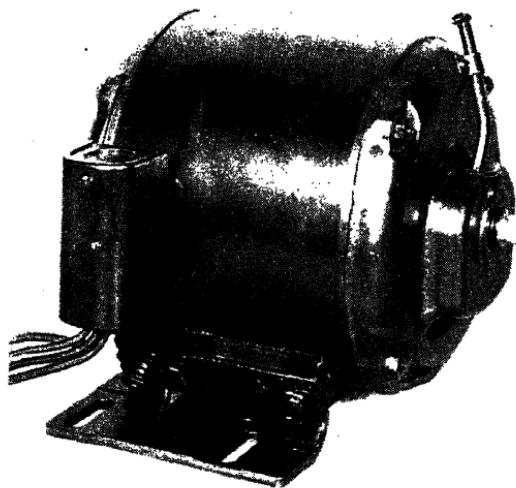


FIG. 107.—Split-phase motor ($\frac{1}{4}$ hp.) with resilient mounting. (Courtesy of Wagner Electric Corporation.)

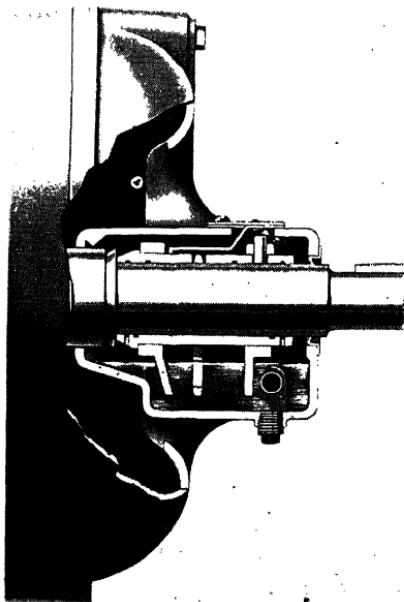


FIG. 108.—Sleeve bearing for electric motors, with lubricating-oil reservoir and oiling ring. (Courtesy of Louis Allis Company.)

overloads up to about 300 per cent rating, at which point stalling occurs.

Small electric motors such as have been discussed in this chapter are regularly furnished by motor manufacturers in three styles of mounting: base, flange and vertical. The vertical mounting has but little application in the oil-burner field, although it is extensively used in the closely allied field of boiler return pumps. However, the vertical-rotary domestic-type burners do use vertical motors; when the cup speed is over 3450 then a series or universal motor is used on both alternating and

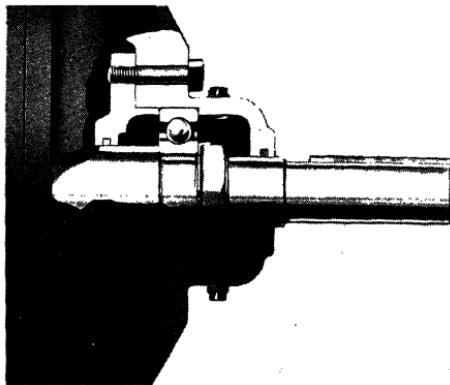


FIG. 109.—Ball bearing for electric motors. (*Courtesy of Louis Allis Company.*)

direct current. The great majority of gun-type burners, both high-pressure and low-pressure atomizing, use flange-mounted motors, with the motor flange usually bolted onto the fan housing. Others use base-mounted motors. Many horizontal-rotary burners are equipped with motors which are really carried in special housings which are extensions of the burner housing and so are part of a special mounting. Both base- and flange-mounted motors can be had with cushion arrangements of springs or rubber to absorb vibration and so aid in quieter burner operation. These cushions support the motor in such manner that the motor is insulated from the base or mounting and any vibration occurring is not transmitted through the cushion.

Conventional fractional horsepower motors are usually furnished with sleeve bearings, as previously mentioned, for quiet-

ness in running. These bearings frequently are provided with an oil reservoir holding as much as a year's supply of lubricating oil. A packing of wool yarn in the bearing bracket serves to bring the oil up to the shaft at the bearing surface. Most motors operate best with about a No. 10 S.A.E. lubricating oil, which is to say a light grade of automobile motor oil.

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CHAPTER X

ELECTRICAL CONTROLS

There are two broad classifications of oil-burner operation—full automatic or intermittent and semiautomatic or continuous. Each comprises many modifications, and a great variety of equipment and apparatus is manufactured for the control of oil burners and related appliances. As a rule, domestic burners are of the full-automatic type. Of recent years, the tendency toward automatic control of commercial burners has been very marked, especially as improvements have been made in both burner design and controls. Some commercial installations are still made for semiautomatic control, however. In the industrial field, semiautomatic control is generally used. In this chapter, only full-automatic controls and methods are discussed, since such controlling is almost exclusively electrical and largely independent of the burner design. Semiautomatic systems for industrial work are discussed in Chap. XV which deals with industrial-plant operation. Semiautomatic controllers are commonly designed for hydraulic operation as well as electrical, and such systems of control are to some extent affected by the burner principle of operation.

Automatic controls in themselves can be grouped as primary controls and operating controls. The primary controls directly govern the starting of the burner whenever burner operation is called for by the operating group. The functions of the primary controls, upon receiving signal from an operating control that burner operation is required, are to energize electrically the motor or motors, if any, and the fuel and air supply systems and the ignition system. Ordinarily, the burner is then in normal operation and fire is being maintained in the boiler or furnace. Should a safe and properly-burning fire not result from this sequence of operations, then the primary controls must again function to shut down the burner so that accumulations of unburned oil cannot form in the firebox. Again, the primary controls are called into action should any interruptions occur to the oil flow

while the burner is running, this time to assure that fuel cannot be sent into a hot firebox until the ignition system is prepared to receive it. Finally, the primary controls usually function also to cut off the ignition, once the burner is running normally and the oil is burning correctly. The primary controls will then keep the burner running until an operating control signals that heat or burner operation is no longer required; the burner then stops.

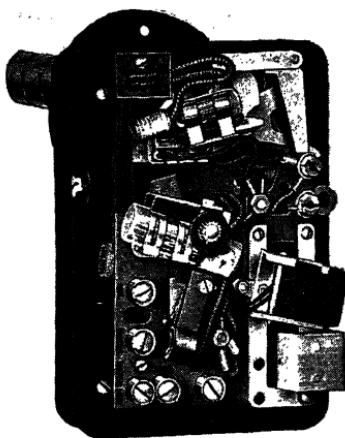


FIG. 110.—Mercoid type KMI Pyratherm-relay and stackswitch combination.

a stackswitch or combustion switch. Since the combination device is less costly, most domestic burners are equipped with such. For the larger motor sizes and different conditions of installation met with by commercial burners, the separate units are more suitable and are generally furnished unless price must be held down to the extreme minimum. Even then, however, the limit of the combination relay-stackswitch is about 1 hp. load.

An oil-burner relay incorporates at least three distinct circuits. The first is the pilot circuit, which is opened and closed as required through the operating controls. In closing, the pilot circuit usually energizes a magnetic switch which in turn closes the main circuit to the motor, fuel system and ignition. Finally, there is the safety circuit, actuated through the stackswitch or equivalent, which interrupts the main circuit in the event of flame failure. The pilot circuit often employs low-voltage current supplied from a transformer. The main circuit, of course, must

RELAYS

Essentially then the primary controls serve two purposes: electrical switching and detection of combustion. Sometimes these two purposes are accomplished in a single device known as a relay-stackswitch combination, or relay-combustion switch combination, and sometimes the electrical switching is handled entirely in one device, then called the relay, while the detection of fire in the firebox is then handled by

be line voltage. The safety circuit can employ either line or low voltage, the former predominating. In the cheaper types of relays, two of these three functions may be combined into a single circuit by carrying the main circuit through the operating controls. But the three circuits are fairly well defined in the average case.

In the usual sequence arranged for in oil-burner relays, the closing of the thermostat, or other operating control switch, causes the relay to close the circuits to the motor, fuel supply

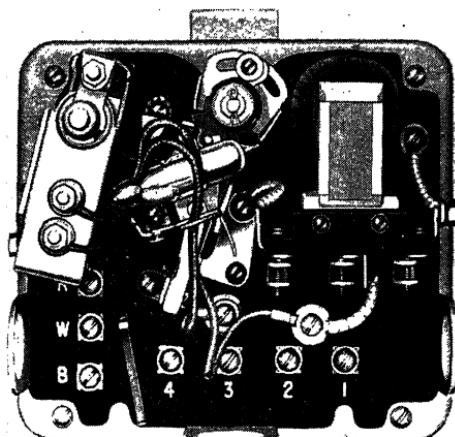


FIG. 111.—Minneapolis-Honeywell type R-117 combination relay and stackswitch.

and ignition. At the same time a thermal element, which is part of a warp switch or other mechanical device for opening an electrical circuit from a heat-actuated stimulus, is placed into circuit. Normally, fuel is delivered into the firebox and is ignited; the resulting fire or hot gases then actuate the stackswitch or combustion control, which then closes its *hot* circuit. In so doing, the stackswitch shunts out the thermal element which has been heating up from the moment the burner started, and also shuts off the ignition. However, should ignition fail, or should the stackswitch not close the *hot* circuit for any other reason, then the continued flow of current through the thermal element will soon cause the warp or other heat-actuated switch to open the main circuit and thus shut down the burner. In the vernacular, the burner is then said to be *off on safety*. Or, should ignition

occur correctly at the time the burner starts, with the stackswitch taking the hot position, only to have the fire die out later for lack of fuel, then the stackswitch assumes the *cold* position, opening the *hot* circuit. This reverse action of the stackswitch may do one or more of a number of things. It may immediately throw the burner off on safety. Or it may, after a scavenger period, bring on the ignition again and at the same time place back into

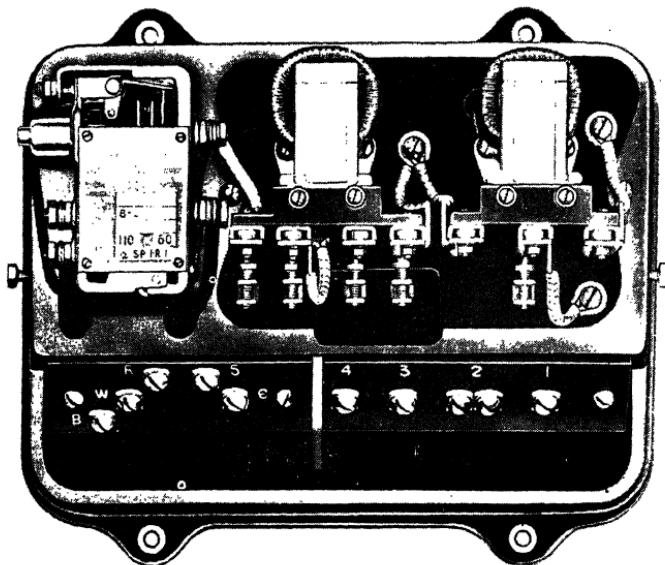


FIG. 112.—Minneapolis-Honeywell type R-114-2 relay, for wall or remote mounting.

the circuit the thermal element in order that it may commence heating and so actuate the warp or other interrupting switch. If now the fuel still does not resume flow, then in due time the thermal element will cause the burner to throw off on safety.

When the sequence of operations is substantially that just described, then the system is referred to as automatic control with intermittent ignition, and is designated by Underwriters' Laboratories as a Group II system. Should the ignition remain on when the stackswitch assumes the hot position, then the ignition is continuous and Underwriters' Laboratories give it the designation of a Group I system.

STACKSWITCHES

Stackswitches or combustion safety switches usually are made sensitive to flame by the action of a helix or coil of bimetal, which is arranged to impart a rotary or linear motion to a shaft which in turn communicates with the electrical switching devices. The bimetal will change its shape or size or position in response to temperature changes, and so the presence or absence of flame is detected. Or, the heat-sensitive element may be a metal diaphragm, instead of a bimetal strip. The bimetal apparatus is usually intended for use in flue passages and breechings where



FIG. 113.—Minneapolis-Honeywell protectostat, a radiant-heat combustion safety switch.

hot gases from the firebox will cause the switch to assume the hot position. When the fire stops burning the cooler gases, drawn up the chimney, in turn move the stackswitch mechanism back to cold or starting position. The metal diaphragm *sights* the fire, and expands from radiant heat. It must therefore be located in a position permitting it to point directly at the fire, usually on the fire door or directly alongside of the nozzle of the burner. Either type of safety, bimetal or disk, will rapidly lose responsiveness to heat when coated with soot. Fortunately, the failure of the combustion safety devices to respond to burner operation because of a soot deposit will cause burners to stop operation, so that hazard of faulty operation is at least obviated.

Sooted coils and diaphragms are frequent sources of service calls to oil-burner dealers.

The coils and diaphragms of combustion safeties are manufactured in varying compositions of metals having different ranges of temperature sensitivity. It is thus possible to choose metals having characteristics most suitable to any given set of conditions as encountered on any particular installation. Manufacturers of such apparatus will furnish specifications on their products, including the factors of temperature ranges available and recommended for various applications.

Typical relays, circuits, combustion safeties and methods of mounting in installation are illustrated on these pages. It is not possible to illustrate all types of all manufacturers; in general principle, all conform to the theories here described. Commercial products are continuously being refined and improved, so that from time to time modifications are made in assembly and design, but as yet no radically new principles or theories of relay and combustion-switch construction have appeared or been announced which indicate that the practice over the past ten years will be rendered obsolete. It is by no means improbable, however, that such will not occur, for in some respects present practice does leave room for improvement, especially as regards complete dependability of operation over long periods of time and as regards freedom from the need for mechanical service by the owner or by a dealer.

THERMOSTATS

In domestic installations of oil burners, the burner is usually started and stopped in response to signals from the room thermostat, which is also a heat-responsive device but constructed to a much more sensitive scale than stackswitches. In the latter, temperature changes of many hundreds of degrees are met with; in the former, 2° of temperature change is considered a fairly large differential, while modern practice in thermostat design aims to hold limits of sensitivity within $\frac{1}{4}^{\circ}$. Another sharp difference between room thermostats and stackswitches lies in the fact that heat from the fire is applied instantly to the flame-detecting instrument, but heat introduced into a room comes only gradually into contact with the room temperature regulator. So it becomes apparent that the room thermostat construction

must be designed from two points of view: to respond to extremely small variations of temperature and in some manner to anticipate that the heat affecting it has come to it only gradually and therefore the source (radiator or warm-air register) must have a reserve built up at the instant that response to it occurs. American manufacturers have been very skillful in meeting these requirements, and their products offer remarkable satisfaction in room temperature-control application.

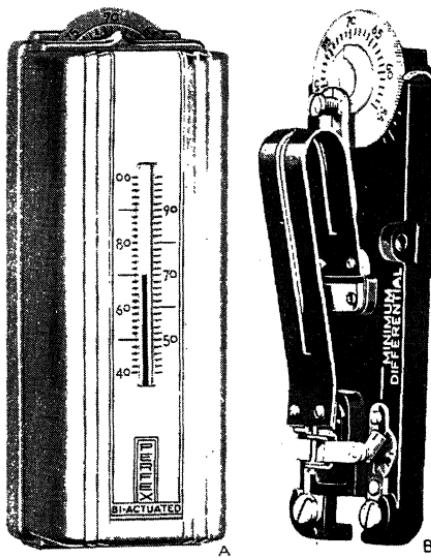


FIG. 114.—Perfex room thermostat.

The most commonly used heat-sensitive medium today is the bimetal strip, although in the past, fluid-filled bellows and metal diaphragms were also extensively used and to some extent still are used. Delicate construction of bimetal arrangements and fine balancing of contact mechanisms give to thermostats close sensitivity to temperature, and ingenious wiring schemes, which incorporate small heating elements in the thermostats themselves, solve the problem of overcoming lag which would otherwise permit overheating by prolonged burner periods of operation. One commercial form of "heat-anticipating thermostat" employs an electrical circuit whereby a definite increase in room temperature must occur before the artificial heat is applied. This prevents the control point from dropping as the weather becomes

severe and insures better heat distribution within the building. A well constructed thermostat also is protected against chattering which might arise from momentary contact caused by vibration of building walls on which the thermostats are mounted.

Alternation of day and night temperatures, when such is desired, can be accomplished in one of two ways. The first is to

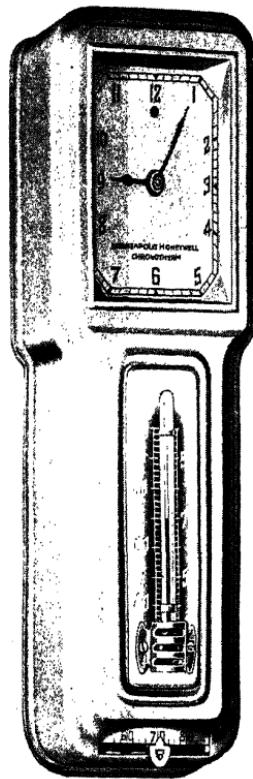


FIG. 115.—Minneapolis-Honeywell chronotherm, an electric-clock room thermostat for day and night temperature alternation, with heat-anticipating device.

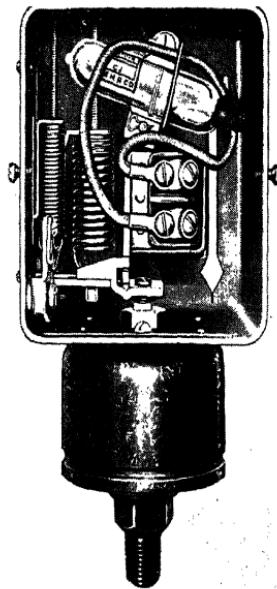


FIG. 116.—Minneapolis-Honeywell pressurestat, or steam-boiler limit-control switch.

build a clock mechanism into the thermostat itself so that the setting can be changed from daytime to nighttime temperature at predetermined hours. The second is to equip the thermostat with two temperature-actuated switches, one of which can be set for the desired daytime temperature and the other for the nighttime, together with a time switch, located in basement or any other convenient spot, which will successively place the daytime and nighttime circuits into control of the burner at

predetermined hours. Usually, regardless of which method is used, the clock mechanism is electrically driven, so that no winding is required. These synchronous clock movements, however, operate only on alternating current. When direct current only is available, then the second method, or dual thermostat, must be used, and the clock can be driven by a direct-current motor which is arranged to wind the spring movement of the clock. Or, of course, a manually wound clock movement can be employed in conjunction with any of the methods just described.

LIMIT CONTROLS

Just as important as room temperature controls are the limit controls which are used to prevent building up of excess pressure or temperature in the heating plant. Included in this group are: pressurestats which limit steam pressures, vaporstats which limit vapor pressures, aquastats which limit water temperatures, and airstats which limit air temperatures, as well as a few others of special nature and application. As a matter of fact, these types of controls are often used as original controls as well as limiting controls, in those cases where thermostats for room temperature are not or cannot be used. Hence each of these devices is usually made in two or more styles, differing in the degree of accuracy possible in their ranges of operation and closeness of differential.

Pressurestats are actuated by either a Bourdon tube, similar to pressure gauges, or a bellows. The transmission of boiler pressure through these elements causes the tipping or tilting of a mercury tube or else moves other electrical contacts. These instruments should always be installed with a trap of some sort to lock a seal of air between the pressurestat and the boiler, which arrangement permits of the transmission of boiler pressure through the air seal into the instrument but prevents the entrance of steam itself. The simplest sort of trap can be made by merely forming a piece of pipe into a loop to form a *pigtail*. These pigtails, which are commonly used, tend to unwind when under pressure, although, of course, the motion is very slight. Nevertheless, in mounting, the instruments should always be turned to a position in reference to the loop so that the ensuing motion from the heating and cooling of the loop will have least effect.

upon the switch. With mercury tubes, this position is with the tube perpendicular to the plane of the loop.

Pressurestats are necessarily made in a wide variety of pressure ranges. For ordinary steam-heating plants, of the gravity type, a pressurestat range of 1 to 5 lb., with a differential setting of as close as 1 lb., is sufficiently accurate if the instrument is used as a limit control only. If, however, the pressurestat is to be the principal operating control rather than, or entirely in lieu of, a

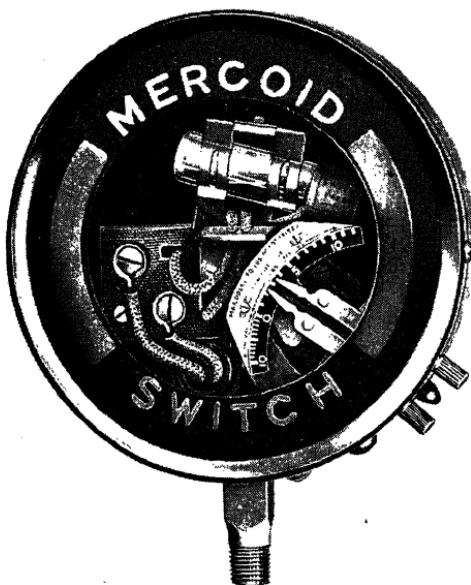


FIG. 117.—Mercoid pressurestat, operating in either pressure or vacuum range, adjustable within close differential.

thermostat, then a differential as low as $\frac{1}{2}$ lb. is desirable, and the cutout pressure should be readily adjustable to suit weather conditions, by a simple and accessible screw or other device. For an operating control on a vapor-heating system, the pressurestat (then called a vaporstat) must be more sensitive, to give a differential of a few ounces pressures. In the case of vacuum-heating plants, it is desirable that the instrument be capable of dropping down into pressures in the vacuum range, *i.e.*, below atmospheric. On the other hand, for high-pressure steam work, pressurestats, of

course, are available in standard stock ranges up to 150 lb. pressure, and on special order can be had for higher pressures.

Aquastats and airstats, since they both regulate or are affected by fluid temperatures, are similar in construction. Both have elements which must be inserted into the boiler, pipe, duct, furnace or other apparatus containing the heated water or air, which elements transmit the effect of expansion or contraction with changes in temperature back to the electrical switching arrange-

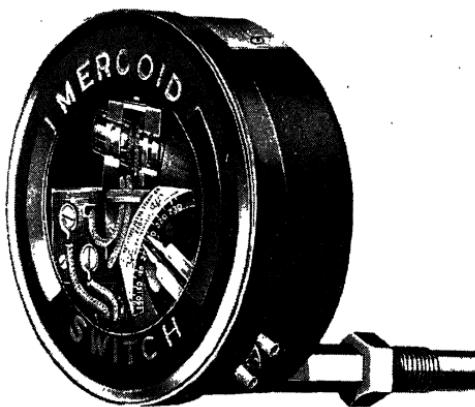


FIG. 118.—Mercoid immersion-type aquastat.

ment of the device. As is true also of pressurestats, the differential sensitivity can be had either fine or coarse, depending on the nature of use to which the controls are to be put. For limit purposes, the less sensitive controls may, of course, be used and are less expensive; when used as direct-operating controls, aquastats and airstats should be selected with finer differential and closer accuracy of setting, the additional cost usually being justified by the results required.

Aquastats are also available for surface mounting as a high-limit control and are used by being strapped or clamped onto the surface of a pipe or boiler, in which position the temperature element receives its heat by transfer through the back plate of the housing which encloses the rest of the mechanism. The element in this type is often a bimetal strip, which actuates the switch in a manner similar to a room thermostat. In this type the differential is not usually adjustable, but the cutout point may be

altered by changing the position of a pointer or lever. Surface aquastats permit of simple and rapid installation, since the need for draining of pipe lines or boilers is obviated.

HEATING SYSTEM CONTROLS

After thermostats and limit controls, the next most commonly used control on a heating system is the low-water safety cutout

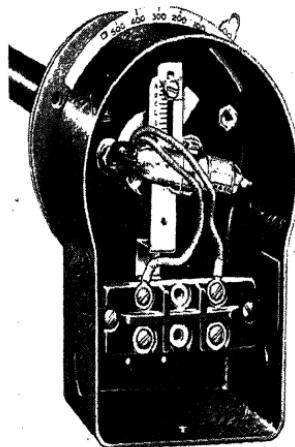


FIG. 119.—Minneapolis-Honeywell Airstat—a warm-air limit-control switch.

that serves to protect the steam boiler against injury from being fired while containing insufficient water. The principal parts of a low-water cutout are: A float chamber which is connected to the boiler in parallel with the water column or gauge glass; a float (generally a copper sphere) connected to the switching mechanism in such manner that switch operation occurs at desired levels of the float in the chamber; and a switching mechanism which can be arranged to either open or close electrical circuits, or both, at low or high position of the float, or both. It is apparent that these devices must have a variety of uses, as they can be employed to interrupt oil-burner circuits or to complete alarm circuits at low-water levels, or to close burner circuit and open alarm circuit at normal water level, or to close another alarm circuit at high-water level. Boiler return-pump control can also be secured from such float controllers, and direct water feed, into boiler from water service line, can also be had by using the float to regulate a water valve.

Of recent years the refinements of methods for heating both buildings and hot water have to a considerable extent been assisted by the use of aquastats and airstats. Thus, in building heating systems employing hot water as the heating medium, forced circulating systems have the marked advantages over gravity circulating systems of giving much quicker heating results and of operating with much smaller pipe sizes and radiator sizes. But the correct and logical control of such systems is

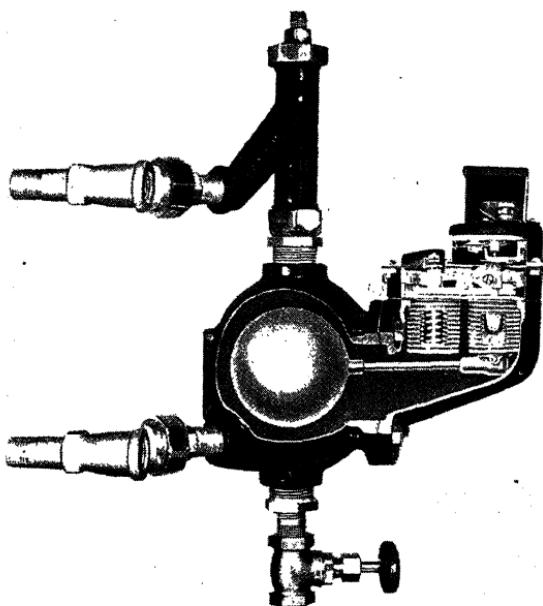


FIG. 120.—McDonell and Miller combination pressurestat and low-water cutout.

accomplished through a combination of aquastats strategically employed to give various ranges of hot-water temperatures during mild weather and severe weather and during periods between thermostat signals of heat. This is to say, the room thermostat merely controls the circulating pump and thereby regulates the temperature of the radiators according as the rooms require heat. In turn, burner operation is directly controlled from boiler-water temperature, through aquastats, there being from one to three, depending on the size of the building and complexity of the heating system. Thus one aquastat may control the

burner during the mild weather to maintain water temperatures between 120 to 180°F.; another may have a range of from 160 to 240°F., to be used during severe weather; and a third may serve as a limit switch to guard against any eventualities which might occur through failure of the rest of the control circuit. Similarly, in modern warm-air heating plants, the room thermostat is used to stop and start the circulating fan while the burner is regulated

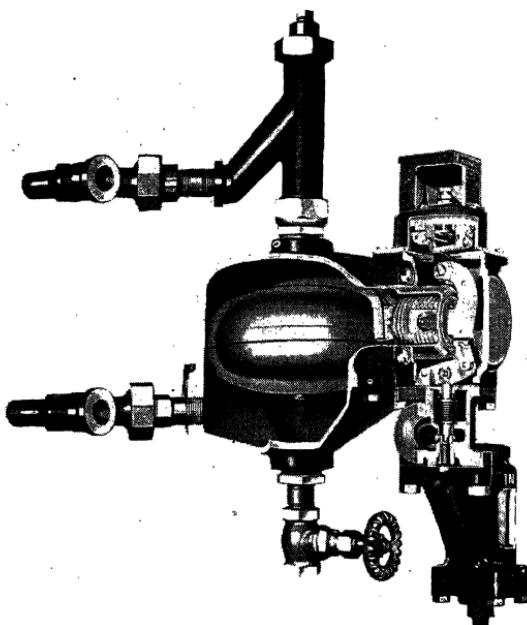


FIG. 121.—McDonell and Miller combination water feeder and low-water cutout.

from an ainstat located to operate from furnace-bonnet temperature. Finally, when domestic hot water is being provided from the same boiler that heats the building, an immersion aquastat may be used to maintain boiler-water temperature at a point that will insure adequate domestic hot water independently of the room thermostat which is controlling room temperatures. Wiring schemes for accomplishing various of these results are illustrated on accompanying pages.

Other refinements in heating-plant control are made possible by the use of electrical switching devices. These are discussed in Chap. XX, which deals with control for heating systems.

Since cost of operation is possibly the most important single item relating to a heating system, other than the design of the system itself, the subject is there given separate treatment and is presented together with the latest theories on heating-plant control.

ELECTRICAL RATING

All electrical controls must be classified according to their electrical rating or ability to carry current. Broadly, instruments are said to be either line or low voltage, designating thereby whether the permissible use is for switching full-line voltage at 110 or 220 volts, or whether the intended use is for the lower values of about 10 to 20 volts which are obtained from transformers or other sources, such as dry cells. Another basis for classification, however, is on the type of electrical switching contacts, there again being two possibilities, open contacts and mercury tubes. In the former case, for low voltage, the contacts may be brought together at almost any desired rate of speed and, similarly, separated again at a leisurely rate. However, when open contacts are used on line voltage, the contacts must be brought together and again separated with a sharp and quick action, in order to avoid arcing of current across a narrow gap. In the case of mercury tubes, the design is sometimes such that a clean break of the mercury is permitted by having the tube assume a position of inclination with a snap action so that the mercury can quickly drop away from the position of contact, although a mercury switch is inherently quick acting. Mercury tubes are also made in low- and line-voltage styles, the difference lying in the size of tube, thickness of contacts and amount of mercury. Practically every control instrument is plainly stamped by its manufacturer to show its electrical characteristics and maximum rating on current.

SOLENOID VALVES

Besides the electrical switching devices which have so far been discussed in this chapter, a discussion of electrical controls should logically include mention of solenoid-operated valves. These extremely useful types of apparatus have been used for many years to govern the flow of oil and gas on automatic burners. A solenoid valve consists essentially of an iron plunger which serves as the stem of the valve, surrounded by an electrical coil or

solenoid which, when energized by current, raises the plunger to open the port of the valve. When the coil is deenergized, the plunger falls back into position to close the port. Sometimes a spring holds the plunger against the port for a close contact, and the plunger compresses the spring in rising. In the case of both gas and oil, it is important that the port areas be large enough to permit the desired rate of flow of the liquid or gas when the valve is opened. Frequently, gas valves, when used for oil-burner ignition, have by-pass arrangements, controllable by a screw

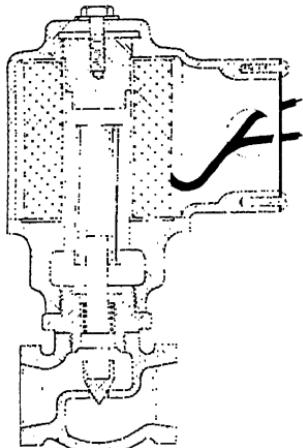


FIG. 122a.—Cross section of solenoid-oil-shutoff valve. (*Courtesy of Automatic Switch Company.*)



FIG. 122b.—Solenoid, or magnetic, oil-shutoff valve. (*Courtesy of Automatic Switch Company.*)

setting, which permits a small flow of gas around the plunger so that a pilot light can be kept burning until the main gas flow occurs whenever the coil is energized. Gas valves seldom are required to open against any significant pressure, but oil valves operate in lines which may carry oil pressures as high as 50 to 100 lb. per sq. in. This load condition must be allowed for when the coil is designed and wound. Both gas- and oil-solenoid valves are frequently designed so that the plungers pull up against the line pressure, so that when the valves are closed there is less likelihood of leakage around the plunger as it rests against the port or seat.

Electric valves larger than $\frac{3}{4}$ -in. pipe size are as a rule operated by an electric motor rather than a solenoid coil. Motorized valves are, of course, much more costly than solenoid valves, but

where the application calls for a large valve size the expense is usually justified by the nature of the installation. It is possible to secure modulating flow through oil valves, having proper motor equipment, for use in industrial oil-burner installations.

MOTOR STARTERS

Very often, especially with commercial and industrial oil burners, the motor load is not carried by the primary burner

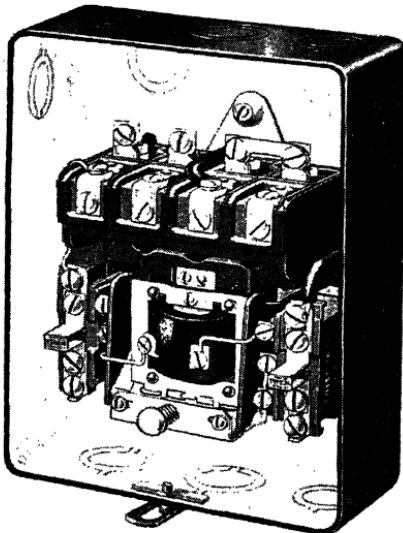


FIG. 123.—Allen-Bradley solenoid motor starter with no-voltage release and overload relays.

control directly, but is independently carried by a motor starter which is operated from a pilot circuit under the control of the primary instrument. In this manner the primary control can be of the same type which is used on smaller burners, and can have the same electric rating (usually maximum of 1 hp.), but through the agency of a magnetic motor starter it can handle the control of much larger electric motors. Also, in special cases where the control circuit is of different voltage or current characteristics than the motor current, this affords a means of relaying signal impulses from one circuit to the other.

The magnetic motor starter itself offers other operating advantages in addition to mere relaying of circuits. The typical magnetic starter is provided with low-voltage release, and overload protection to guard the motor against injury through serious

excesses of current passing through it. Again, in the case of direct-current motors, a dashpot arrangement can be incorporated in the motor starter which will start the motor at low speed and bring it to full speed gradually by means of a starting rheostat or resistance.

Since the overload protection of this equipment is one of its most important features, it is important to consider it in some detail. Most manufacturers utilize the motor current to exert a thermal effect so that when the motor current becomes excessive the thermal effect becomes sufficient to melt a soldered connection thereby releasing a spring arrangement which opens the entire electrical circuit. Hence it is imperative for the proper application of this equipment that the correct motor current under normal operation be known, and that the tripping relay open as soon as that normal current is exceeded by the safe percentage. To accomplish this in practice, the overload relays are so designed that the motor current is carried through a thermal or heating element, where the heating effect takes place. These thermal elements are made in a large variety of sizes or capacities, so that the correct element may be chosen for any particular motor. The normal motor currents for various types and sizes of motors are shown in the following table. Values shown are amperes at full load.

TABLE XXV.—FULL-LOAD MOTOR CURRENTS, IN AMPERES

Motor size, hp.	1 Phase 110 v.	1 Phase 220 v.	2 Phase 220 v.	3 Phase 220 v.	3 Phase 440 v.	Direct current 115 v.	Direct current 230 v.	Direct current 550 v.
$\frac{1}{8}$	2.7	1.4						
$\frac{1}{6}$	3.0	1.5	2.7	1.4	
$\frac{1}{4}$	4	2	1	1.0	0.6			
$\frac{1}{3}$	5.2	2.6	1.1				
$\frac{1}{2}$	7.5	3.75	1.9	2.2	1.1	4.5	2.3	1
$\frac{3}{4}$	10	5	2.8	3.1	1.6	6.5	3.3	1.4
1	12.5	6.1	3.5	4	2	8	4	1.7
$1\frac{1}{2}$	18	9	4.5	5.1	2.6	12.5	6	2.6
2	24	12	5.5	6.25	3.2	16	8	3.4
3	34	17	7.9	9	4.5	24	12	5
5	55	28	13.1	15	7.5	38	19	8.2
$7\frac{1}{2}$	80	40	19.3	22	11	58	29	12
10	105	53	25.4	29	14.5	75	38	16

When magnetic motor starters with overload relays are employed in oil-burner work, there is introduced a factor which

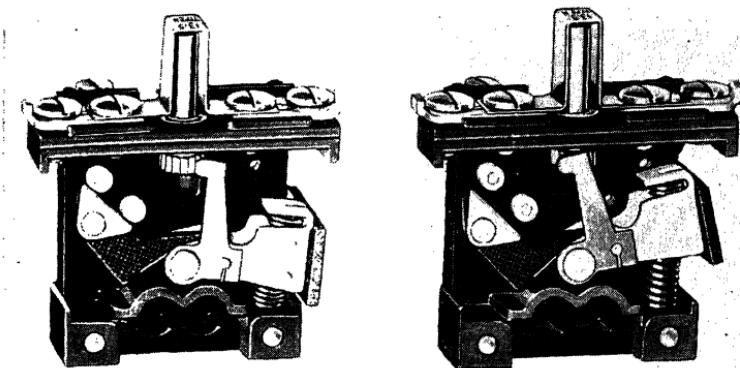


FIG. 124.—Overload relays for Allen-Bradley solenoid motor starter. Left, closed or normal position; right, open or tripped position.

is not present in the average case where similar equipment is used, *i.e.*, an abnormal temperature in the boiler room. Necessarily, the control apparatus of an oil burner must be located in fairly close proximity to the burner, and usually the room temperature (or ambient temperature, as it is sometimes more scientifically called) is higher than is customary in ordinary locations. The thermal relay of the starter will be affected by this room temperature, and other things being equal, the thermal device will trip at a lower current value than would normally be the case. To avoid unnecessary interruptions of the oil-burner operation, the thermal element selected must therefore be somewhat higher than would be chosen were the room temperature

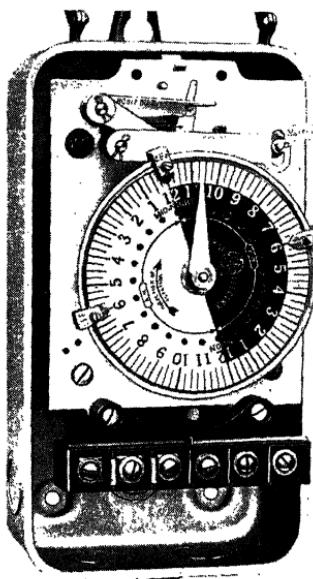


FIG. 125.—General Electric time switch with electric-clock drive.

normal. Care must be exercised that the thermal element is not

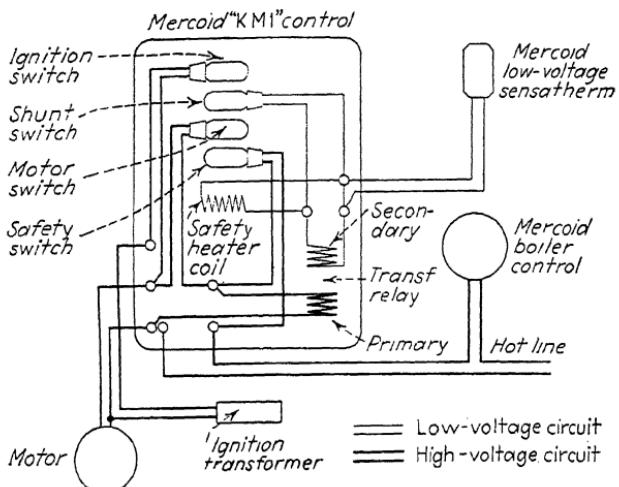


FIG. 126a.—Wiring diagram for Mercoid KMI Pyratherm, for domestic burners.

selected for too high a current value, however, for in that case the entire overload protective apparatus will lose its usefulness. For if the required current value needed to trip the relay is so great that the motor burns out before the relay trips, then nothing has been gained by providing the motor safeguard.

In the case of less expensive installations of burners with single-phase motors, the overload protection is omitted, and the magnetic starter is utilized only to handle the motor current. Sometimes also, magnetic switches of the same type must be used to control current supplies to other devices than motors, as, for example, electric oil preheaters, or where it is desired to operate a multiplicity of circuits from a single pilot circuit, as, when a thermostat or pressurestat is

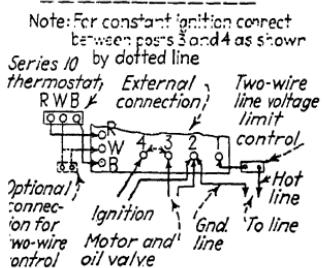


FIG. 126b.—Wiring diagram for Minneapolis-Honeywell R-117 relay, for domestic burners.

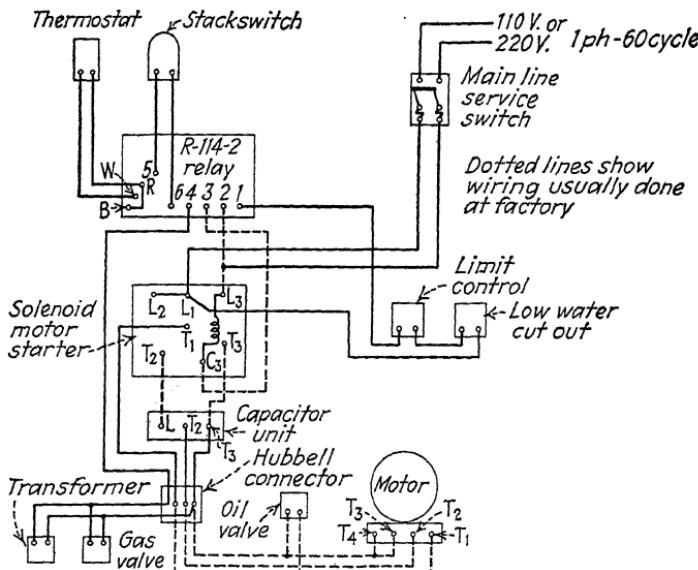


FIG. 127.—Wiring diagram for single-phase motor on commercial or industrial oil burner: capacitor motor, Minneapolis-Honeywell R-114-2 relay, solenoid motor starter, and electric-gas ignition.

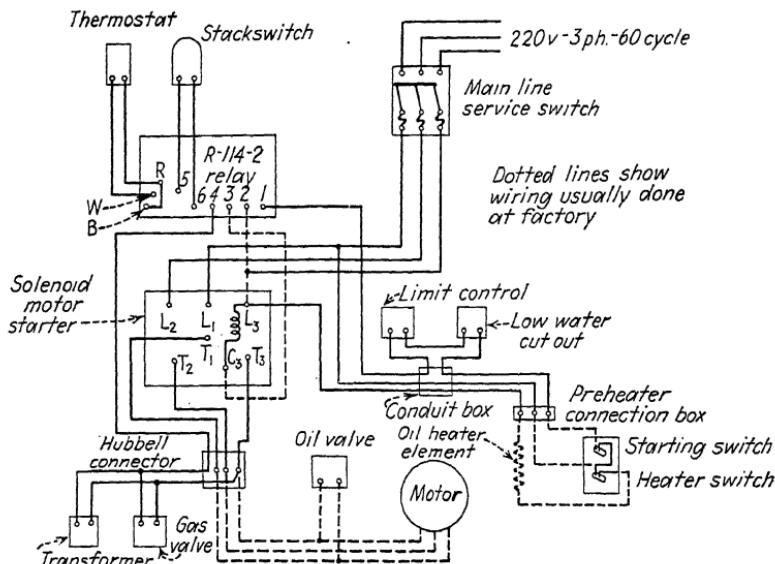


FIG. 128.—Wiring diagram for three-phase circuit on commercial or industrial oil burner: Minneapolis-Honeywell R-114-2 relay, solenoid motor starter, electric-gas ignition, and electric oil heater with the mostatic regulator and lockout circuit.

controlling the starting and stopping of two or more burners or motors.

Magnetic-motor starters are frequently employed for manual starting and stopping of motors. Many commercial and industrial oil burners are so equipped. In such cases, a push-button start-and-stop station is provided, either in the cover of the starter itself or in a separate metal box adjacent to the starter, to which the box is connected by a short conduit nipple to carry the

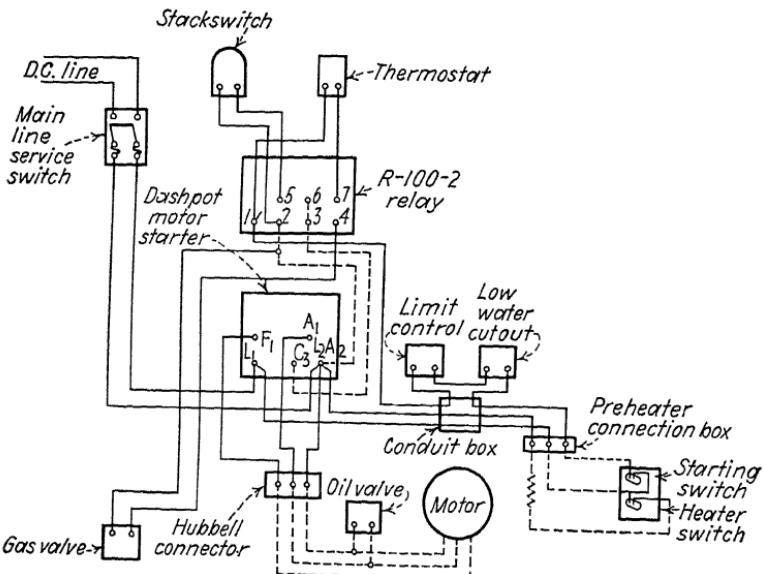


FIG. 129.—Wiring diagram for direct-current circuit on commercial or industrial oil burner: Minneapolis-Honeywell R-100-2 relay, dashpot-type solenoid motor starter, expanding-gas ignition, and electric oil heater with thermostatic regulator and lockout circuit.

control wires. The start button, when pushed in, completes the magnetic-coil circuit, which then pulls in the armature which makes the motor contacts. The armature also carries an extra contact which shunts out the start button, thereby keeping the coil energized after the start button is released. When the stop button is pushed in, the coil circuit is broken. The armature then falls away and opens both the motor circuit and the shunt contact around the start button. The tripping of the overload relay also opens the coil circuit, causing the armature to fall to the off position. Once the overload relay has tripped, it must be manually reset.

When the magnetic-coil circuit is completed through a primary burner control, instead of by the push-button station, the functioning of the starter is exactly the same as above described.

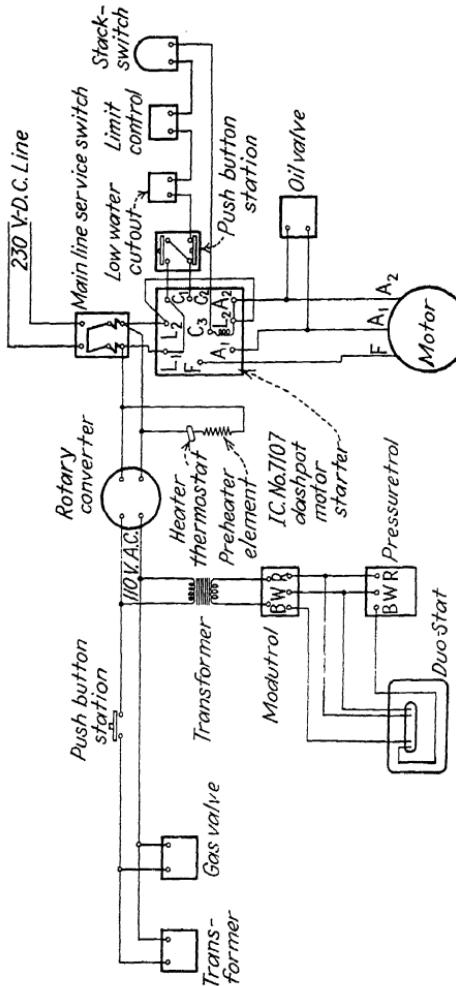


FIG. 130.—Wiring diagram for direct-current circuit on semi-automatic commercial or industrial oil burner having manually operated ignition system and modulating-fire control.

Just as in the case of oil-burner controls, every magnetic-motor starter has definite maximum electrical capacity. This load-carrying limit is usually plainly marked somewhere on the device. The manufacturers' maximum current specifications should never be exceeded. Usually, these mechanisms are tested and approved by Underwriters' Laboratories, who also set load limits for them.

CHAPTER XI

IGNITION METHODS

Once the burner has brought the fuel oil into the firebox in the correct physical condition and mixed with the correct amount of air for combustion, it is necessary that some arrangement then function to set fire to, or ignite, the oil vapor or spray. With the exception of certain types of continuously burning domestic burners which employ an oil pilot light to keep the fire maintained between periods of major burner operation, ignition must occur each time that an automatic burner starts in response to signal from some sort of operating control. It is true also in the case of manually started burners, that it is necessary to light up the oil when first starting up, but this is usually no more complicated a procedure than inserting a burning torch or other burning material into the path of the oil and of then turning on the oil. But in automatic burners the ignition is always an important function of the mechanism and sometimes is one of the principal problems of the oil-burner design.

There are two media for effecting ignition of fuel oil: gas flame and electric spark. Sometimes the two are combined together in what is then termed electric-gas ignition. In the early days of automatic oil burning, gas was the original and sole means of ignition, and very often, in those early models, the same gas flame was used in vaporizing the fuel as well as igniting it. But there soon developed two serious objections to gas as an ignition means: (1) its cost and (2) its restricted distribution to metropolitan districts. With the development of atomizing nozzles there no longer existed the need for the gas flame as a vaporizing means, so the electric spark was adopted almost exclusively as ignition for domestic work. But in automatic commercial burners, using heavier grades of fuel, electric-spark ignition is not suitable, and gas must be employed. Frequently, to conserve gas consumption, commercial burners are provided with electric-gas ignition, whereby the gas does not burn until the burner is ready to start, at which instant the gas flow commences and is ignited by an electric

spark. The burning gas then lights up the oil flame. Usually, once the oil is burning, both the spark and gas then shut off. Gas is still used for ignition, although really for initiating vaporization, in some types of distillate burners.

Electric ignition is, therefore, by far the most commonly used. The source of the electric current for the high-tension spark is the ignition transformer. These transformers are designed for operation on a primary current identical with that served to the burner's electric motor. The secondary voltages range from 5000 to 15,000, depending on the design of the burner. The most common voltage is 10,000. These transformers are made so that they can operate in continuous service, for those applications where the spark remains on as long as the burner is operating. But the general practice is to shut off the spark as soon as the oil flame becomes established.

Transformers, of course, can operate only on alternating current; in fact, they depend, for the induction of the current in the secondary winding, on the alternation of the current supply in the primary winding. The induced current in the secondary winding will have the same frequency as the primary supply, but the voltage of the induced supply is determined by the ratio of the number of turns of wire in the secondary to the number in the primary.

It follows that transformers cannot be used on direct current. There are various ways of handling the ignition problem on direct-current burners, when spark ignition is required. The most popular method on gun-type burners is to provide a set of collector rings on the direct-current motor from which a small supply of alternating current can be drawn to serve a special transformer (see Chap. IX, Fig. 99). A second method is to install a small rotary converter which will generate the required alternating current at the desired voltage and frequency. This is somewhat more expensive than the collector-ring scheme but must sometimes be used, especially when an additional source of alternating current is required on direct-current installations for the operation of other control apparatus. A third way is to use a spark coil, similar to the type used for automobile ignition. This is probably the least satisfactory (although it is inexpensive), for the reason that spark coils do not give long service on high-voltage primary circuits.

Ignition transformers can be classified as single pole or two pole, the former generally being used in combination with gas in electric-gas ignition burners, while the latter are customarily used for straight electric-ignition burners. Occasionally, however, a single-pole transformer is used for electric ignition. In the single-pole type, one side of the secondary winding is grounded to the casing and hence to the burner, while the other side of the secondary is connected to the electrode. This arrangement then causes a spark to jump from the electrode to any other electrical



FIG. 131.—Single-electrode ignition transformer, 5000-volt secondary. (*Courtesy of Dongan Electric Manufacturing Company.*)

conductor which is brought sufficiently close to it so that the electric potential of the secondary winding can overcome the resistance of the air gap. The electrode is, of course, brought into juxtaposition with the other conductor which is to complete the circuit to ground in such manner that the spark gap occurs in the correct position to effect ignition. Where the single-pole transformer is used it is necessary to insulate only one electrode and the connections to it.

The two-pole transformer, which is by far the most commonly used since most domestic burners employ this type for straight electric ignition, also has a ground connection, called the midpoint ground. This arrangement really makes the secondary consist of two windings, each having one side jointly grounded and having the other sides serving their respective electrodes, which two electrodes form the spark gap for the system. In the

typical case, each of the two windings generates 5000 volts and the electrodes are at that potential with respect to ground. But

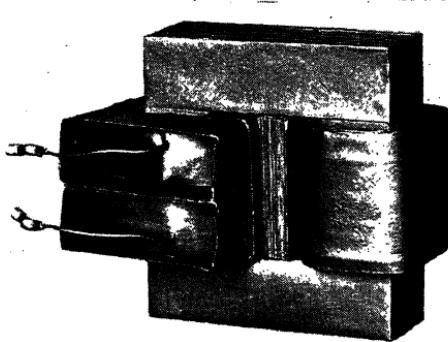


FIG. 132.—Assembled coils and core for shielded-ignition transformer, high voltage mid-point grounded, 10,000-volt secondary. (*Courtesy of General Electric Company.*)

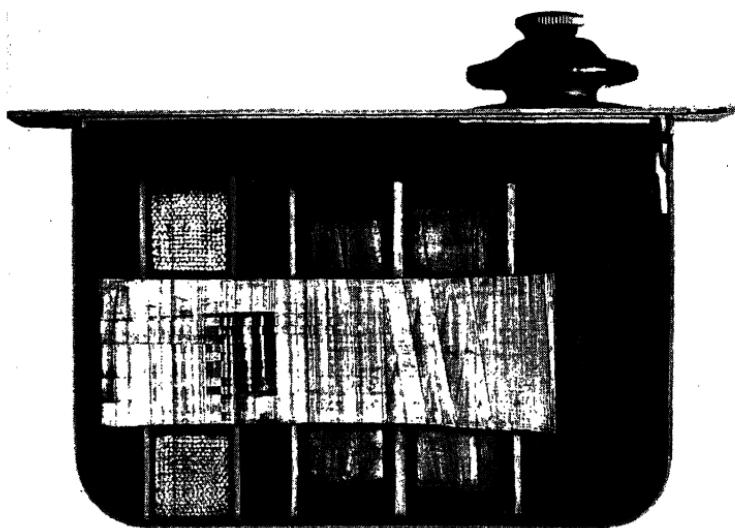


FIG. 133.—Section through ignition transformer. (*Courtesy of Webster Electric Company.*)

between the two electrodes themselves there exists a 10,000-volt potential difference. For a 12,000-volt transformer, each side has a 6000-volt winding; for 15,000 volts each side consists of a

7500-volt coil. The advantage of the mid-point ground is that each electrode and connection thereto need be insulated against only half of the actual transformer voltage. Also, should someone, in working on the burner or transformer while it is in operation, accidentally touch an exposed part of one side, the shock received will be from only half of the full voltage.

Wherever possible, the high-potential wire which must be used to connect the transformer terminal to the electrode should be

enclosed entirely within a metal body, such as the air tube of the burner. This avoids possible radio disturbances, and also safeguards the wire from possible short circuits, due to worn

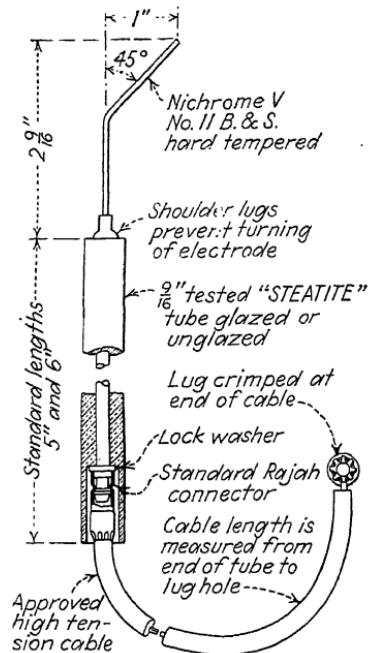


FIG. 134a.—Details of ignition electrode, insulator, and high-tension cable. Connector joining cable and electrode is recessed into insulator tube. (*Courtesy of Dielectric Products Corporation.*)

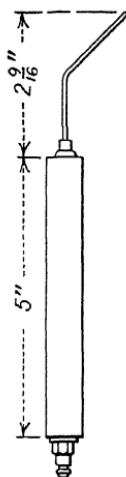
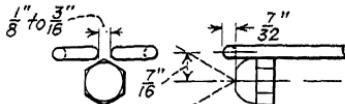


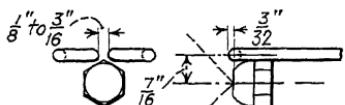
FIG. 134b.
Ignition electrode and insulator; connector is exposed.

insulation, with consequent hazard of shock to person. The transformer used should also be of the shielded type, which is the designation used to refer to a transformer so manufactured that it causes no radio interference. Until recently manufacturers of ignition transformers resorted to condensers to avoid radio complications, but the condenser, because of certain inherent disadvantages, has been abandoned in favor of shielding.

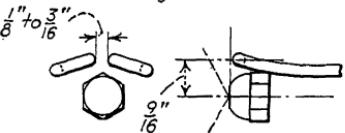
Present practice in electrode construction is toward the use of nichrome-steel wire with rounded ends at the gap, rather than pointed ends. The pointed end wears away rapidly, and so makes the gap increase in size, which of course ultimately results in ignition failure. For the standard 10,000-volt transformer used to ignite No. 3 fuel oil, the gap should be $\frac{1}{8}$ to $\frac{3}{16}$ in. in length. For higher voltages the gap can be proportionately longer; for 5000 volts the gap cannot be much more than $\frac{1}{16}$ in. The accompanying illustrations show proper methods of placing the electrodes with respect to the nozzle tip. It should be noted that on straight electric-ignition burners, it is not necessary to place the electrode in the oil spray, but that the gap can be slightly back of the edge of the spray. The air stream passing over the gap causes the spark to assume a curved shape, thereby actually blowing the spark into the oil spray. Should the electrode tips actually reach into the oil spray, carbonization will



Correct setting with 45° nozzle



Correct setting with 60° nozzle



Correct setting with 80° nozzle

FIG. 135.—Correct setting of ignition-electrode points for gun-type burners.

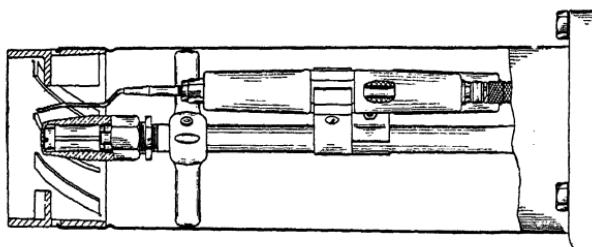


FIG. 136.—Section through end of air tube, showing positions of oil feed tube, nozzle, air diffuser, and electrode.

inevitably occur and, of course, will soon bridge across the gap and cause ignition failure.

When gas only is used for ignition, the gas flow is generally controlled by a solenoid gas valve which embodies a by-pass around the plunger so that a small amount of gas may continue to flow

when the plunger is down. This small gas flow maintains a pilot light, permitting the main flow of gas, when the plunger opens, to enlarge this pilot into a large gas flame that will serve to ignite the oil. The actual amount of gas flowing through the by-pass is controlled by a screw adjustment. In general, the by-pass adjustment should be set at the lowest gas flow that will permit the pilot to burn without danger of its blowing out from the air currents which occur in the firebox. The pilot should be enclosed in some manner to shield it from these air currents. If difficulty

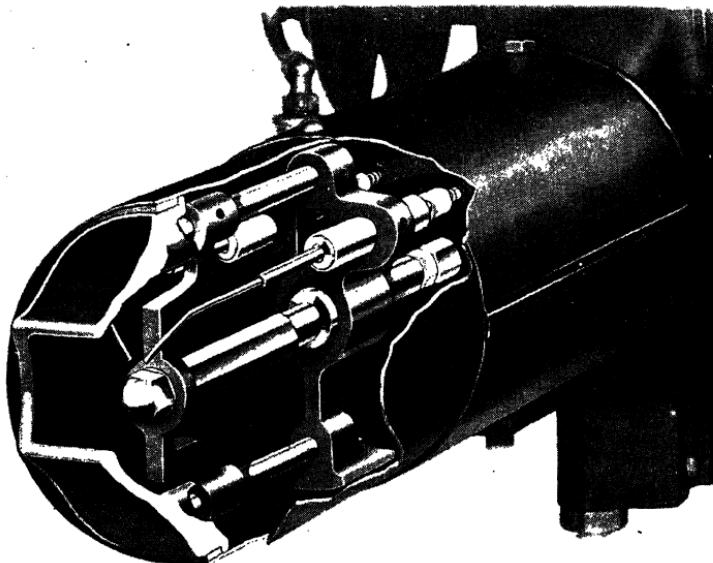


FIG. 137.—Combustion head of pressure-atomizing burner. (*Courtesy of Harvey-Whipple Inc.*)

is encountered in maintaining the pilot with only a reasonable flow of gas, it may be due to a characteristic of the quality of the gas being served by the utility company. In such cases the service or engineering department of the gas company can usually give information as to the arrangement and setting of the pilot to overcome this difficulty.

The electric-gas ignition method has largely supplanted the expanding-gas flame. There are two objections to the use of gas only for ignition: first, the cost of the gas burned in the pilot, between periods of expanded gas flame and secondly, the tendency of pilots to blow out if sudden changes or disturbances of

draft occur in the firebox. The electric-gas ignition method is more positive and less costly in gas consumption. The slight additional investment required in a transformer and other simple accessories to the electric-gas ignitor are well justified in view of the advantages accruing from its use.

It sometimes happens that it is desired to operate commercial or industrial oil burners under automatic control in places where gas is not served by a utility company. In such cases a solution may readily be had by resorting to the use of gas stored in steel tanks—the so-called bottled gas. Since the consumption of gas can be held very low by employing electric-gas ignition, a tank of gas will hold out over a fairly long period. The most common gas found in use for this purpose is propane, but if propane cannot be had in a given locality, ordinary acetylene as it comes in tanks for use on the acetylene torch can be devoted to this use very satisfactorily.

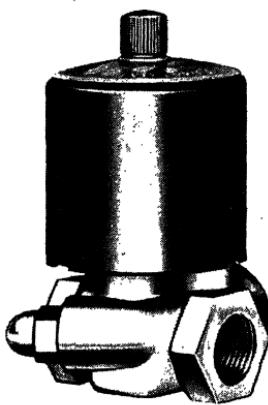


FIG. 138.—Solenoid gas valve with adjustable by-pass
(Courtesy of Minneapolis Honeywell Regulator Company.)

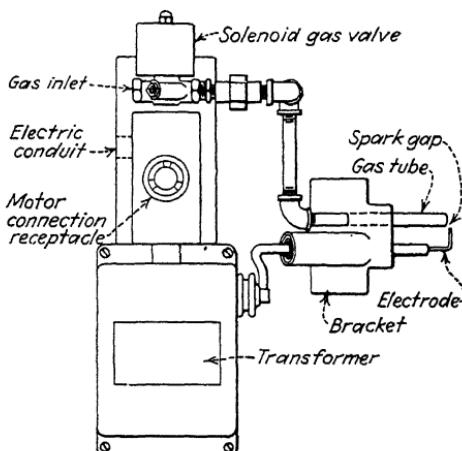


FIG. 139.—Electric-gas ignition assembly. (Courtesy of Ace Engineering Company.)

Where electric-gas ignition is employed, it is not generally feasible to enclose the high-tension wire, but on the other hand

this wire can be made very short, possibly not more than 3 or 4 in. in total length, and the ignition system should certainly be constructed in such manner. As mentioned above, 5000 volts is ample for igniting a gas flame. A single-pole transformer is the best style to use, for then the gas pipe itself can be the other electrode.

The design of the ignition system on an automatic oil burner merits the best thought and attention which can be devoted to it. Probably the most common source of service on such installations is the failure of ignition. The designer can do much to assure continuous operation to the user and freedom of service to the dealer by producing a dependable method of ignition on his burner.

CHAPTER XII

FUEL-OIL PUMPS AND PUMPING

Fuel oil must be mechanically handled in practically all phases of its varied applications and uses. In fact its adaptability to such mechanical handling constitutes one of its major advantages, for moving or conveying of this fuel can thus be accomplished through the agency of pumps and piping, eliminating manual handling such as often is necessary with solid fuels. The physical transmission of oil through pumps and piping involves a number of subordinate operations: straining, regulation of pressure, regulation of quantity or volume of oil delivered to burners and maintenance of correct fuel-oil temperatures. Hence a fuel-oil pumping system logically begins at the storage tank and ends at the burners. Between these two extremes is the equipment which establishes and maintains the fluid flow: pumps, piping, fittings, valves, heaters, strainers, pressure regulators, metering and measuring devices. Each contributes to the common cause or purpose of delivering oil where and as needed and in the correct physical condition; each therefore should be considered as part of a system, and the system to be an entity rather than an assemblage of discrete parts. This chapter is devoted to discussion of pumps and conduits of oil flow, and the next chapter treats of the remaining elements of the fuel-oil transmission system.

Fuel-oil pumps can be generally classified into two divisions—external- and internal-gear types. Although there are other types of pumps, of course, than the gear types, such are seldom used for fuel oil. Thus centrifugal pumps are widely used in industry for moving water and other nonviscous liquids, but are seldom applied to viscous pumping. Reciprocating pumps were originally the only pumps used in the petroleum industry; refineries and pipe lines of forty years ago used them exclusively. But today the use of reciprocating pumps is confined to high-pressure steam plants, and there the principal advantage is in the ease of developing high oil pressures when required. So for most prac-

tical purposes, a discussion of fuel-oil pumps can be confined to a discussion of gear pumps.

Both external- and internal-gear pumps are encountered in a wide and diverse range of forms. Their designers try to achieve

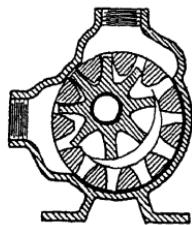


FIG. 140.—Popular form of internal-gear pump. Rotation is reversible; if clockwise, upper port is suction and side port is discharge; if counter-clockwise, the ports reverse.

two ends—to provide for transporting oil and to fashion gear surfaces and forms that will avoid congestion within the pump at those points where oil is forced through small passages or apertures. In general, the internal-gear type can be constructed at a lower cost than the equivalent external-gear type, by reason of there being less casing and less machine work needed. Hence, for small oil burners, especially domestics, internal-gear pumps are most popular. Indeed, the internal type serves very well for a great deal of commercial and industrial work as well, but when equipment is desired that will yield the utmost in operating performance, and when first cost is not a primary consideration, then external gear-type pumps are usually selected.

Figure 140, in section, shows one of the commoner types of internal-gear forms used commercially, while the form shown in Fig. 141 is the same pump in a frame extensively used in domestic burners. A few commercial forms of external-gear pumps are illustrated in Fig. 142. Here may be noted the extensive variation of design which is intended to overcome or minimize the occurrence of congested areas of fluid within the pump. In the less expensive models of external gear, and almost always in the internal gear, one of the gears is power driven while the other is turned by mesh with the driven gear and is called the idler gear. In the more costly types, especially of external-gear pumps, both gears are power driven, being interconnected by a timing-gear mechanism.

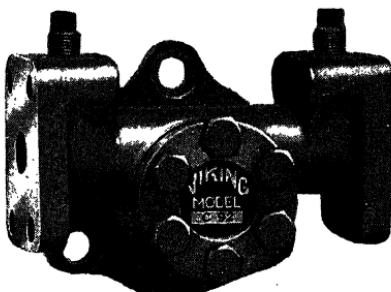


FIG. 141.—Internal-gear pump with flanged connections. (Courtesy of Viking Pump Company.)

Too often in industry, the selection of a pump is predicated upon first cost, little thought being given to such matters as

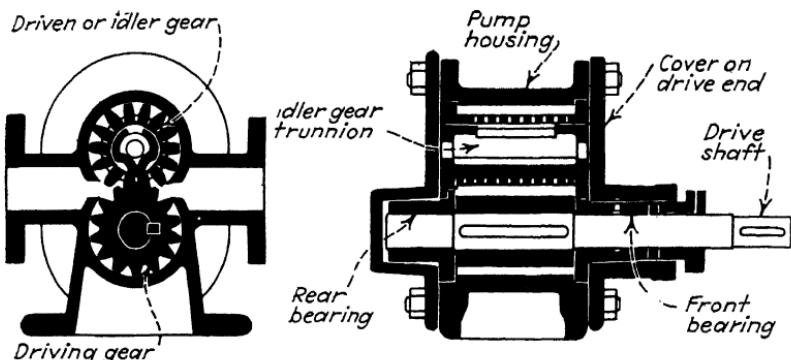


FIG. 142a.—External-gear pump. (*Courtesy of Schutte and Koerting Company.*)

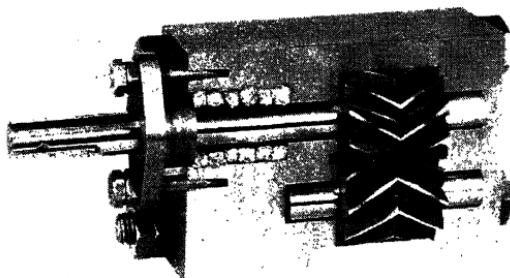


FIG. 142b.—External-gear pump. (*Courtesy of Northern Pump Company.*)

maintenance, ultimate life and, more especially, to power consumption. As a rule, the tendency for power consumption to increase out of proportion to good economy is greater as the required pressure increases, which is another way of saying that the feature of power consumption becomes increasingly important as the developed or required oil-pressure increases.

The speed at which rotary pumps should be operated is related to the viscosity of the oil being pumped. Table XXVI shows a general relationship between these variables. Economy in power consumption will be heavily influenced by the speed, so that excessive speeds should be guarded against by providing

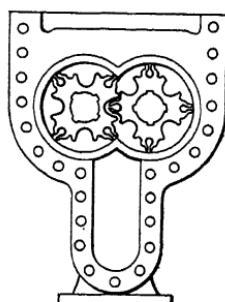


FIG. 142c.—External-gear pump.

pumps which will develop adequate capacity and pressures at nominal speeds, as indicated by the chart.

TABLE XXVI.—SPEED-VISCOSITY TABLE, SHOWING RELATION BETWEEN VISCOSITY, PUMP SPEED, PUMP TYPE AND PUMP CAPACITY

Pump type	External-gear type						Internal-gear type						
	35	50	100	200	300	600	5	20	50	90	200	300	450
Viscosity Saybolt universal sec.							Maximum pump speed, r.p.m.						
50	1750	1150	850	500	400	300	1750	1150	400	390	375	300	260
100	1750	1150	850	500	400	300	1750	1150	400	390	375	300	260
200	1750	1150	850	500	400	300	1750	1150	400	390	375	300	260
300	1750	1150	850	500	400	300	1750	1150	400	390	375	300	260
400	1750	1150	850	500	400	300	1750	1150	400	390	375	300	260
500	1750	1150	850	500	400	300	1750	1150	400	390	375	300	260
600	1740	1125	830	490	395	295	1740	1125	395	385	370	295	256
700	1715	1100	810	480	390	290	1715	1100	390	380	365	290	252
800	1690	1075	790	470	385	285	1690	1075	385	375	360	285	248
900	1665	1050	770	460	380	280	1665	1050	380	370	355	280	244
1000	1640	1025	750	450	375	275	1640	1025	375	365	350	275	240
2000	1580	990	725	430	355	260	1580	990	355	350	334	260	230
3000	1520	955	700	410	335	245	1520	955	335	330	317	245	215
4000	1460	915	675	390	315	230	1460	915	315	310	300	230	200
5000	1400	875	650	370	295	215	1400	875	295	290	280	215	185
6000	1340	835	625	350	275	200	1340	835	275	270	260	200	170
7000	1280	795	600	330	255	185	1280	795	255	250	240	185	155
8000	1220	755	575	310	235	170	1220	755	235	230	220	170	140
9000	1160	715	550	290	215	155	1160	715	215	210	200	155	125
10000	1100	675	525	270	195	140	1100	675	195	190	180	140	110
20000	1000	600	500	250	175	120	1000	600	175	170	170	120	100
30000	900	525	450	230	170	110	900	525	170	165	160	110	90
40000	800	450	400	210	160	100	800	450	165	160	150	100	80

In the great majority of oil-burner installations on heating plants, fuel is drawn from the storage up to the pump by suction, thence to be delivered to the nozzle under some pressure, which may be low or high according to the burner type. Perhaps the most important feature of the suction line, aside from its being

large enough, is that it be absolutely tight and free from air leaks. Regardless of the pump type used, air leaking into the suction line universally produces ill effects in the pump-and-flow regulating mechanism: knocking and hammering, wheezing and sputtering, chattering and clattering, all occur in the pump and valves by reason of air drawn into the suction line. In turn, these mean excessive pump wear, uneven pressure and fluctuating delivery. A tight suction line is the start of a good oil-burner installation—a leaky one is its finish.

GENERAL HYDRAULIC THEORY

While its velocity is below the critical point, the flow of fuel oil in a pipe is streamline; when the critical velocity has been



FIG. 143.—Types of liquid flow. In streamlined, a film of liquid clings to the pipe surface, and smooths the path of flow. In turbulent, eddy currents prevent film formation and the friction against pipe surface is greater.

exceeded, the flow is turbulent. Usually there is a small range of velocity values where the flow changes gradually from the one type of flow to the other—this is designated as the critical range.

Streamline flow is typical viscous flow. It may be likened to the way in which the several concentric cylinders of a telescope slide one within the other; but in the liquid flow there would be a great many cylinders of extreme thinness. In fact, the definition of viscosity is based upon this conception of telescoping cylinders, and it may be stated as the tendency of one layer of the liquid to resist moving at the same speed as the adjacent concentric layer, thus giving rise to a slipping effect. We need not concern ourselves with it here, but simply note in passing, that the measure of viscosity is the measure of the stress developed in this slipping of the molecules over each other in the liquid body.

Figure 143 is an illustration of the two types of flow. In the case of streamline flow it shows how the extreme outer layer serves as a barrier between the main body of the liquid and the rough walls of the pipe. The laws governing streamline flow are pred-

iated upon the establishment of this outer layer or cylinder which smooths the path of flow. In turbulent flow there no longer exists this stationary film of oil on the pipe wall. Many eddy currents are set up in the liquid body—currents moving in directions other than the direction of the main stream itself. Hence the roughness of the pipe exerts far more influence upon the stream than in streamline flow, and a different set of conditions prevail which require modification of the laws of flow.

As is true of all hydraulic calculations of flow through pipe lines, the most important of the factors which affect a pipe's carrying capacity is friction, and in the case of oil piping the selection of the friction factor with suitable accuracy is the principal problem involved.

There is no short-cut method for securing a friction factor by relatively simple means—even for only an approximate value. On the other hand, the means for enabling the selection of the proper factor are well developed, and with the aid of the accompanying formulas and charts the amount of calculating is reduced to a minimum.

Most design problems in oil pumping involve the factors of oil viscosity, rate of flow, pump capacity, pressure drop and power consumption. All of these quantities, except the first, are usually expressed in units either directly suitable for use in engineering formulas or readily convertible to such units. But in the case of oil viscosity, the commonly used term of Saybolt universal seconds must be converted into other units, called centipoises, for use in these calculations.¹

The centipoise is the unit of absolute viscosity, and it is derived from the expression for kinematic viscosity. The latter term is secured directly from Saybolt universal seconds and is merely the absolute viscosity in centipoises divided by the specific gravity. The unit of kinematic viscosity is the centistoke; the relationship between centistokes, k , and Saybolt universal seconds, t , is expressed by the following formulas.

When t is 100 or less:

$$k = 0.226t - \frac{195}{t} \quad (1)$$

¹ Other commercial scales of viscosity are also used, principally Saybolt furol in the United States. Saybolt furol and other scales can be converted to Saybolt universal, by Fig. 7, Chap. II.

TABLE XXVII.—FOR CONVERSION OF SAYBOLT SECONDS TO CENTIPOISES
AND OF A.P.I. DEGREES TO SPECIFIC GRAVITY

Saybolt sec. t	Kinematic viscosity Z/s	A.P.I. Degrees											
		4	6	8	10	12	14	15	16	17	18	19	20
		Specific gravity											
		1.04	1.03	1.01	1.00	0.986	0.973	0.966	0.959	0.953	0.947	0.940	0.934
50	7.40	7.70	7.62	7.47	7.40	7.30	7.20	7.14	7.10	7.05	7.0	6.95	6.91
55	8.89	9.25	9.15	8.98	8.89	8.76	8.65	8.58	8.52	8.46	8.42	8.35	8.30
60	10.33	10.75	10.65	10.45	10.33	10.20	10.07	10.00	9.91	9.85	9.78	9.72	9.65
65	11.70	12.18	12.06	11.81	11.70	11.53	11.40	11.30	11.21	11.14	11.08	11.0	10.92
70	13.04	13.58	13.44	13.18	13.04	12.85	12.70	12.60	12.50	12.42	12.35	12.28	12.20
75	14.35	14.91	14.78	14.50	14.35	14.15	13.98	13.85	13.77	13.65	13.59	13.50	13.40
80	15.66	16.30	16.12	15.80	15.66	15.42	15.25	15.10	15.0	14.90	14.82	14.71	14.62
85	16.90	17.60	17.40	17.08	16.90	16.67	16.46	16.30	16.20	16.10	16.0	15.90	15.20
90	18.19	18.90	18.72	18.38	18.19	17.91	17.70	17.55	17.45	17.32	17.20	17.10	17.0
95	19.40	20.20	20.00	19.60	19.40	19.12	18.90	18.72	18.60	18.48	18.35	18.25	18.12
100	20.65	21.50	21.25	20.85	20.65	20.40	20.10	19.93	19.80	19.65	19.52	19.40	19.30
125	26.4	27.5	27.2	26.6	26.4	26.1	25.7	25.5	25.3	25.2	25.0	24.8	24.6
150	32.1	33.4	33.0	32.4	32.1	31.7	31.3	31.0	30.8	30.6	30.4	30.2	30.0
175	37.7	39.2	38.8	38.1	37.7	37.2	36.7	36.4	36.2	35.9	35.7	35.4	35.2
200	43.3	45.0	44.5	43.8	43.3	42.7	42.1	41.8	41.5	41.3	41.0	40.7	40.5
225	48.9	50.8	50.3	49.4	48.9	48.2	47.5	47.1	46.9	46.6	46.3	46.0	45.8
250	54.5	56.6	56.1	55.0	54.5	53.8	53.1	52.6	52.3	52.0	51.6	51.3	50.9
275	60.0	62.5	61.8	60.6	60.0	59.2	58.5	58.0	57.5	57.2	56.9	56.5	56.0
300	65.5	68.0	67.5	66.1	65.5	64.6	63.8	63.3	62.9	62.5	62.0	61.5	61.1
325	71.1	74.0	73.3	71.9	71.1	70.2	69.4	68.8	68.3	67.9	67.4	67.0	66.5
350	76.1	79.1	78.5	77.0	76.1	75.2	74.1	73.6	73.0	72.5	72.1	71.8	71.2
375	82.1	85.4	84.5	83.0	82.1	81.0	80.0	79.4	78.9	78.4	78.0	77.3	76.7
400	87.7	91.2	90.3	88.5	87.7	86.5	85.5	84.6	84.1	83.6	83.0	82.5	82.0
425	93.2	96.8	96.0	94.2	93.2	92.0	90.6	90.0	89.3	88.9	88.3	87.6	87.0
450	98.7	102.6	101.5	99.8	98.7	97.4	96.0	95.4	94.7	94.0	93.5	92.8	92.1
475	104.2	108.2	107.1	105.0	104.2	102.8	101.5	100.9	99.8	99.3	98.6	97.9	97.2
500	109.7	114.0	113.0	110.8	109.7	108.0	106.8	106.0	105.0	104.5	104.0	103.0	102.0
525	115.5	120	119	117	116	114	112	111.5	111	110	109.5	108.5	108
550	121	126	125	122	121	120	117.7	117.0	116	115.5	114.5	114.0	113
575	126	131	130	127	126	124	122.5	121.5	121	120	119	118.5	118
600	132	137	136	133	132	130	128.5	127.5	127	126	125	124	123
625	137.5	143	142	139	138	136	134	133	132	131	130	129	128
650	143	149	147	144	143	141	139	138	137	136	135	134	133
675	148.5	154	153	150	149	147	145	144	143	142	141	140	139
700	154	160	159	156	154	152	150	149	148	147	146	145	144
725	159.5	166	164	161	160	157	155	154	153	152	151	150	149
750	165	171	170	167	165	163	161	160	158	157	156	155	154
775	170.5	177	176	172	171	168	166	165	164	162	161	160	159
800	176	185	181	178	176	174	171	170	169	168	167	166	165
825	181.5	188	187	183	182	179	177	175	174	173	172	171	170

TABLE XXVII.—FOR CONVERSION OF SAYBOLT SECONDS TO CENTIPOISES AND OF A.P.I. DEGREES TO SPECIFIC GRAVITY.—(Continued)

Saybolt sec. <i>t</i>	Kinetic vis- cosity <i>Z/s</i>	A.P.I. Degrees											
		4	6	8	10	12	14	15	16	17	18	19	20
		Specific gravity											
		1.04	1.03	1.01	1.00	0.986	0.973	0.966	0.959	0.953	0.947	0.940	0.934
850	187	194	193	189	187	184	182	181	179	178	177	176	175
875	192.5	200	198	194	195	190	188	186	185	184	182	181	180
900	198	206	204	200	198	195	193	191	190	188	187	186	185
925	203.5	211	210	206	204	200	198	196	195	194	193	191	190
950	209	217	215	211	209	206	205	202	200	198	197	196	195
975	214.5	223	222	217	215	212	209	207	206	205	203	202	200
1000	220	229	227	222	220	217	214	212	211	210	208	207	205
1100	242	252	249	244	242	239	236	234	232	231	229	228	236
1200	264	275	272	267	264	260	257	255	253	252	249	248	247
1300	286	297	295	289	286	282	278	276	274	273	271	279	267
1400	308	320	317	311	308	304	300	298	295	294	291	290	288
1500	330	343	340	333	330	326	321	319	316	315	312	310	308
1600	352	366	362	355	352	347	342	340	338	336	333	331	329
1700	374	389	385	388	374	369	364	361	359	356	354	351	349
1800	396	412	408	400	396	391	385	382	380	378	377	372	370
1900	418	435	431	423	418	412	407	404	392	399	396	394	391
2000	440	458	455	444	440	434	427	425	422	418	416	413	411
2500	550	572	567	555	550	542	535	532	527	524	520	517	514
3000	660	686	680	666	660	651	642	638	633	628	624	620	617
3500	770	800	783	777	770	760	750	745	738	733	728	723	719
4000	880	915	906	888	880	869	856	850	844	833	832	827	822
4500	990	1030	1020	999	990	975	964	956	950	943	936	930	925
5000	1100	1140	1130	1110	1100	1086	1070	1063	1055	1050	1040	1030	1028
6000	1320	1370	1360	1330	1320	1302	1285	1275	1270	1260	1250	1240	1230
7000	1540	1600	1590	1560	1540	1520	1500	1490	1480	1470	1460	1450	1440
8000	1760	1830	1810	1780	1760	1736	1710	1700	1690	1680	1665	1655	1645
9000	1980	2060	2040	2000	1980	1954	1930	1910	1900	1890	1870	1860	1850
10000	2200	2290	2270	2220	2200	2170	2140	2120	2110	2095	2080	2070	2050
20000	4400	4580	4530	4440	4400	4340	4270	4250	4220	4180	4160	4130	4110
30000	6600	6860	6800	6660	6600	6510	6420	6380	6330	6280	6240	6200	6170
40000	8800	9150	9060	8880	8800	8690	8560	8500	8440	8330	8320	8270	8220
50000	11000	11400	11350	11100	11000	10860	10700	10630	10550	10500	10400	10300	10280

When t is over 100:

$$k = 0.220t - \frac{135}{t} \quad (2)$$

From the definition of kinematic viscosity, it follows that, if Z be the absolute viscosity in centipoises, and s the specific gravity,

$$Z = ks \quad (3)$$

Table XXVII shows values of kinematic viscosity for values of t ranging from 50 to 50,000, and for values of Z through the same range of t values, as well as for various specific gravities.

If a problem involves a certain pipe size for which the pressure drop under the conditions of flow stated is to be evaluated, then the friction factor may be secured by referring the numerical value of DVs/Z to Fig. 144 or from the equation $f = 0.00207 \cdot \frac{Z}{DVs}$.

If, however, as is more often the case, the problem is to determine pipe size, then a pipe size must be assumed in order that a tentative friction factor may be selected. If the pipe size assumed is later shown to be incorrect, a more accurate pipe size can then be selected for a second trial calculation, with the proper friction factor.

The expression DVs/Z is called the modulus. In it, D is the pipe diameter, in inches, V the average velocity of flow, in feet per second (discharge divided by cross-sectional area of the pipe), s the specific gravity and Z the viscosity in centipoises. It should be noted that all viscosity values must be those for the fluid at the pumping temperature.

V may be quickly secured from Fig. 145, which shows values of V plotted against pipe sizes for various discharge rates, or from the relationship

$$V = \frac{0.404 \times \text{g.p.m.}}{D^2} \quad (4)$$

in which g.p.m. stands for gallons per minute.

It should be noted from Fig. 144 that the numerical value of the modulus determines the nature of the flow; *i.e.*, whether it be streamline or turbulent. For values under 0.122 streamline flow obtains in steel and cast-iron pipe; for drawn-copper and brass

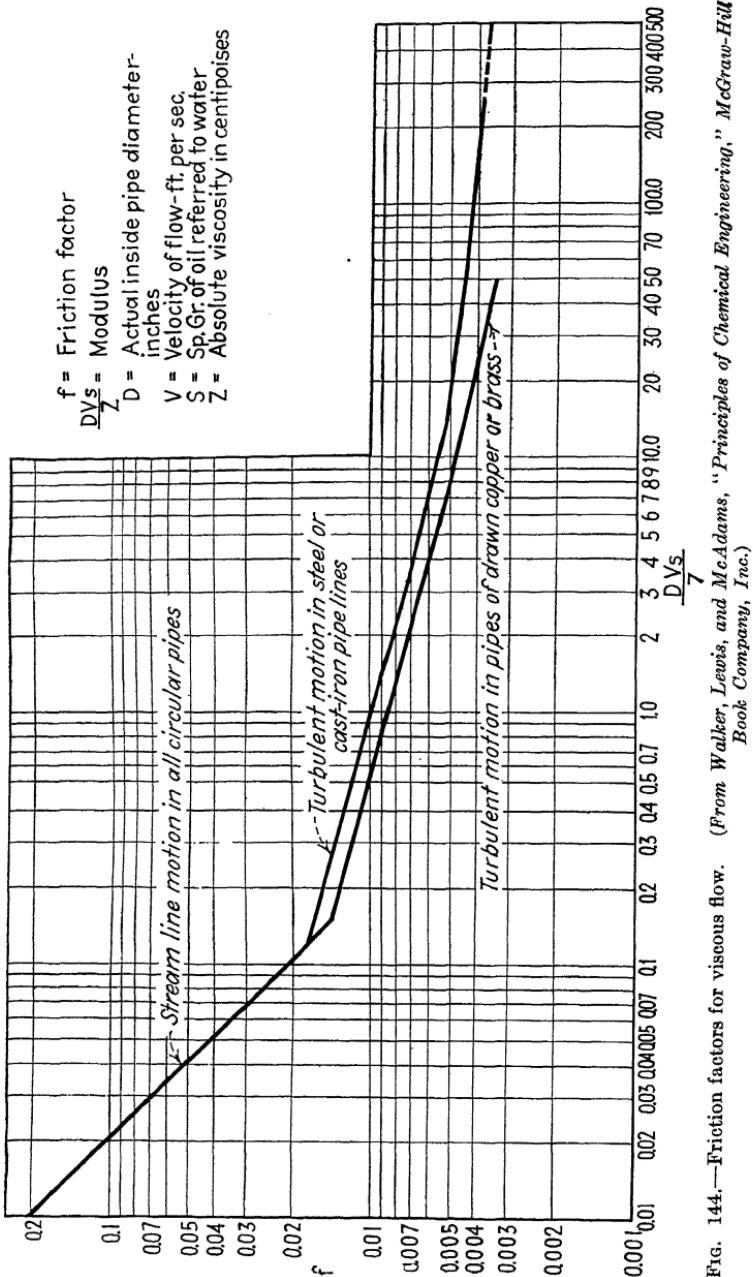


FIG. 144.—Friction factors for viscous flow. (From Walker, Lewis, and McAdams, "Principles of Chemical Engineering," McGraw-Hill Book Company, Inc.)

pipe, the critical point occurs at 0.144. As was pointed out, it is clear that when the stationary film exists on the pipe walls the nature of the wall has less effect upon the friction factor than with turbulent flow.

The friction factor f having been selected from Fig. 144, it is possible to substitute in the general pipe-flow formula to deter-

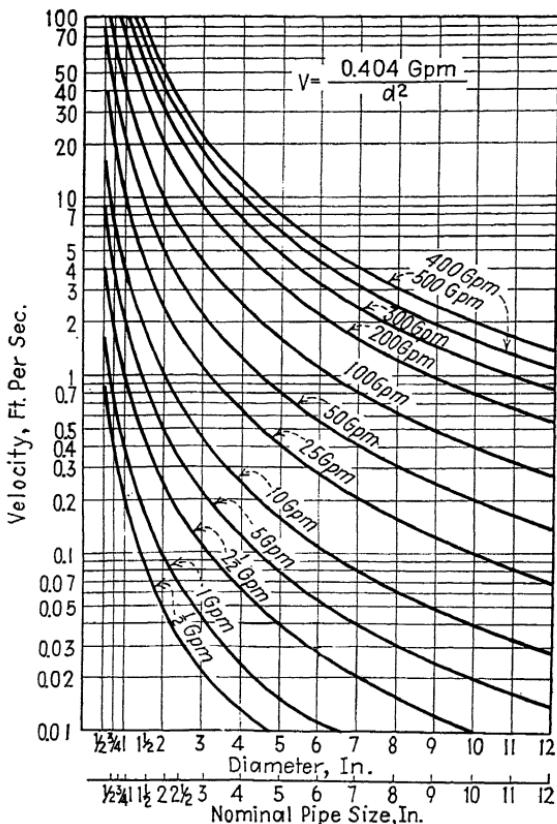


FIG. 145.—Chart showing average velocities of flow through pipes, for various diameters and discharge rates.

mine the pressure drop for the equivalent length of pipe involved in the problem. This same formula holds for both types of flow; the friction factor makes the necessary corrections for whatever variations occur because of the turbulence. In general form, the formula is

$$\frac{2g}{d}$$

in which Δp = pressure drop, lb. per sq. ft.

f = friction factor.

w = density of fluid, lb. per cu. ft.

L = total equivalent length of pipe, ft.

V = velocity, ft. per sec.

g = acceleration due to gravity, 32.2 ft. per sec. per sec.

d = diameter of pipe, ft.

The formula is somewhat inconvenient to use in this form, so the following form is derived from it by using 62.5s for w (s being specific gravity), changing from d , in feet, to D , in inches, and combining all the constants to give Δp , in pounds per square inch:

$$\Delta p = \frac{0.324fsLV^2}{D} \quad (6)$$

The theoretical horsepower required for overcoming the frictional resistance to the flow of oil is expressed as a function of gallons per minute pumped, G , and the pressure drop and specific gravity:

$$\text{Theoretical } hp_f = 0.000582\Delta pGs \quad (7)$$

Since neither the pump nor the motor which drives it operates at 100 per cent efficiency, it is necessary to divide the theoretical horsepower by the product of the pump and motor efficiencies to obtain the actual horsepower developed in overcoming friction through the pipe. Letting M represent motor efficiency and P pump efficiency:

$$hp_f = \frac{0.000582\Delta pGs}{MP} \quad (8)$$

The total horsepower required to transfer a liquid from one point to another is the sum of the power to overcome friction, the power to elevate the liquid from its initial to its final level, and the power to overcome other losses, such as the loss of entrance of the liquid into the pipe and the loss through the pump, etc. This may be expressed as

$$hp_t = hp_f + hp_e + hp_m \quad (9)$$

in which hp_t = total horsepower developed by motor.

hp_f = power required to overcome pipe friction.

hp_e = power required to change liquid level.

hp_m = power required to overcome other losses, such as entrance and pump.

For most practical purposes, hp_m can be taken as 10 per cent of hp_f ; hp_e is determined from the actual vertical distance between the original and final liquid levels, and is numerically equal to $0.00025esG/MP$, e being the difference in liquid level in feet. Hence the complete expression for total horsepower developed can be written as a function of gallons per minute pumped, specific gravity, and e , M and P as above:

$$hp_t = \frac{Gs}{M} [0.00025e] \quad (10)$$

The application of these formulas is best illustrated by working out a few typical problems.

Problem 1.—A railroad tank car holding 10,000 gal. of fuel oil is to be unloaded at a rate of 300 g.p.m. through a pump and a 3-in. pipe line into a vertical storage tank whose capacity is 2,000 gal. per vertical foot. The

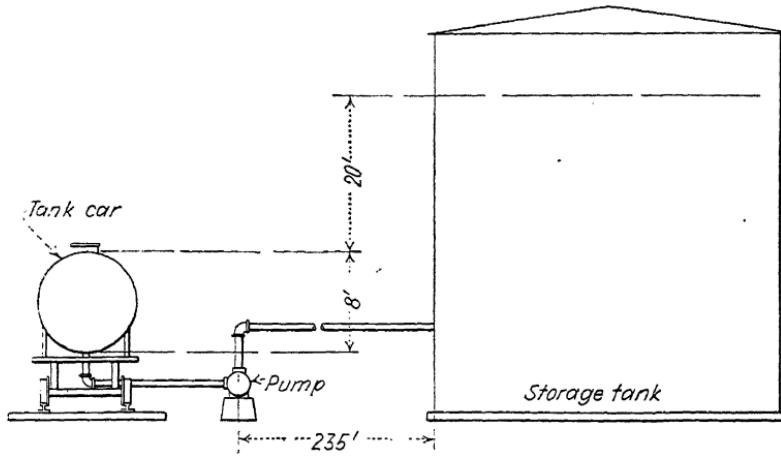


FIG. 146.—Layout for Problem 1.

fuel oil has an A.P.I. gravity of 17 and a viscosity of 900 Saybolt universal sec. at the pumping temperature. The pump efficiency is 55 per cent, and the motor efficiency 75 per cent. Find the cost with electrical power at the rate of 3 cts. per kw.-hr. for unloading the car under the conditions stated, the various elevations and distances being as shown in Fig. 146.

Solution.—From tables, 17 degrees A.P.I. translates into 0.95 for s . From Eqs. (2) and (3), Z equals 188. By Fig. 145 or Eq. (4), V is 13. Hence the modulus, DVs/Z equals $(3.07 \times 13 \times 0.95)/188 = 0.203$. By

referring to Fig. 144 it is noted, first, that the flow is turbulent, and second, that the corresponding value of f is 0.015. Substituting in Eq. (6),

$$\Delta p = 0.324 \frac{0.015 \times 0.95 \times 235 \times 13 \times 13}{3.07} \quad 59.5 \text{ lb. per sq. in.}$$

At the start of pumping, the difference in liquid level is 20 ft. At the end, the tank level is 5 ft. higher, and the car level 8 ft. lower than at the start, making a total difference of 33 ft. in liquid level when the pumping is finished. The average difference in liquid level is therefore 26.5 ft., and this value should be used for e . The power expended in pumping is, by Eq. (10),

$$h_{p.i} = \frac{300 \times 0.95}{0.75 \times 0.55} (0.00064 \times 59.5 + 0.00025 \times 26.5) \\ = 30.6$$

The total time for unloading the car is $10,000/300$ or 33.3 min. The energy expended for unloading is $33.3/60 \times 30.6 \times 0.746 = 12.7 \text{ kw.-hr.}$ At 3 cts. per kw.-hr., the power cost is 38.1 cts.

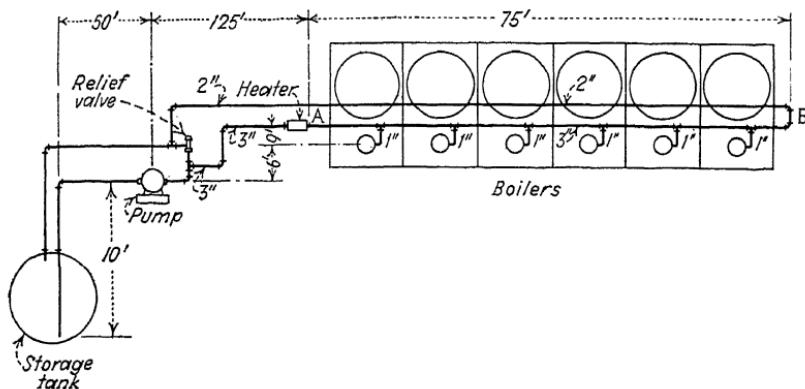


FIG. 147.—Layout for Problem 2.

Problem 2.—Fuel oil is to be drawn from storage and delivered to a battery of six oil-fired boilers, each developing 1000 boiler hp. Oil in the storage tank has an A.P.I. gravity of 8 and a Saybolt universal viscosity of 3000 at 100°F., which temperature is maintained in the tank by heating coils. At the point *A* shown in Fig. 147, a preheater raises the oil temperature and reduces the oil viscosity to 415 Saybolt sec., the gravity changing to 14 A.P.I. The fuel has a calorific value of 156,000 B.t.u. per gal., and is burned at an efficiency of 70 per cent. Hydraulic losses through fittings are taken as 30 diameters for ell and 60 for tees. If the pump efficiency is 45 per cent and the amount of oil returned to storage is one-third, calculate the size of electric motor required.

Solution.—Since a boiler horsepower is defined as 33,500 B.t.u. per hr., the total fuel burned is $(33,500 \times 6000)/(0.7 \times 156,000)$, or 1840 g.p.h. Add one-third this amount for oil by-passed back to storage, and the quan-

ity of oil pumped is 2450 g.p.h., or 41 g.p.m. Hence the rates of flow through the various sizes of pipe, and the respective velocities, are: 3-in. pipe, 41 g.p.m. and 1.76 ft. per sec.; 2-in. pipe, 10 g.p.m. and 0.95 ft. per sec.; 1-in. pipe, 5.1 g.p.m. and 1.87 ft. per sec.

The initial physical condition of the oil gives $s = 1.01$ and $Z = 667$. At the preheater, near the boilers, these values become, respectively, 0.97 and 88.3. The total equivalent length of 3-in. pipe between the tank and the pump, allowing for 1 ell is 67.5 ft. From the pump to point *A*, allowing for 3 ells and 1 tee, the equivalent pipe length is 177.5 ft. The equivalent pipe length from the heater to point *B*, including the 2 ells and 6 tees, is 180 ft. The length of return line is figured back to the tank at 290 ft., and the length of each feed line from the header down to each burner regulating valve is 9 ft., making a total of 54 ft. for the six burners.

The corresponding moduli are 0.0082, 0.0598, 0.0218 and 0.0217. The respective friction factors are accordingly 0.252, 0.0345, 0.096 and 0.096. It is now necessary to perform five separate calculations of pressure drop with Eq. (6).

$$\begin{aligned}\Delta p_1 &= 0.324 \frac{0.252 \times 1.01 \times 67.5 \times 1.76 \times 1.76}{3.07} = 5.7 \\ &= 0.324 \frac{0.252 \times 1.01 \times 177.5 \times 1.76 \times 1.76}{3.07} = 14.7 \\ \Delta p_3 &= 0.324 \frac{0.0845 \times 0.97 \times 180 \times 1.76 \times 1.76}{3.07} = 1.9 \\ \Delta p_4 &= 0.324 \frac{0.096 \times 0.97 \times 290 \times 0.96 \times 0.96}{2.07} = 3.9 \\ \Delta p_5 &= 0.324 \frac{0.096 \times 0.97 \times 54 \times 1.87 \times 1.87}{1.05} = 5.4\end{aligned}$$

$$\text{Total pressure drop} = 31.6 \text{ lb. per sq. in.}$$

Calculations of the power losses in these five sections of piping are made by five applications of Eq. (10), except that the efficiency factor for the pump is introduced at the end. Taking the above pressure drops in succession:

$$\begin{aligned}hp_{.1} &= 41 \times 1.01 \times (0.00064 \times 5.7 + 0.00025 \times 10) = 0.252 \\ hp_{.2} &= 41 \times 1.01 \times (0.00064 \times 14.7 + 0.00025 \times 15) = 0.545 \\ hp_{.3} &= 41 \times 0.97 \times (0.00064 \times 1.9 + 0.00025 \times 0) = 0.050 \\ hp_{.4} &= 10 \times 0.97 \times (0.00064 \times 3.9 + 0.00025 \times -19) = -0.023 \\ hp_{.5} &= 5.1 \times 0.97 \times (0.00064 \times 5.4 + 0.00025 \times 6) = 0.025\end{aligned}$$

$$\text{Total theoretical hp. required} = 0.849$$

Dividing by the pump efficiency gives the motor size required as 0.849/0.45 or 1.88 hp. Of course, this in turn will have to be divided by the efficiency of the motor to be used to give the actual motor capacity required. For this particular application, it would be wise to select a 3-hp. motor.

Problem 3.—A bulk storage plant is to be built on a waterway dock, oil to be unloaded from barges and pumped into 250,000-gal. storage tanks, each 40 ft. in diameter and 26.5 ft. high. There will be six tanks, one for each of the six grades of oil handled. As shown in Fig. 148, one pump and pipe

line will serve two tanks, the three sets handling, respectively, 6 and 5, 4 and 3, 2 and 1, grades of oil. The pump for Grades 6 and 5 will have a capacity of 500 g.p.m., the other two will pump 700 g.p.m. The maximum allowable pumping head against which the pumps must work is 75 lb. per

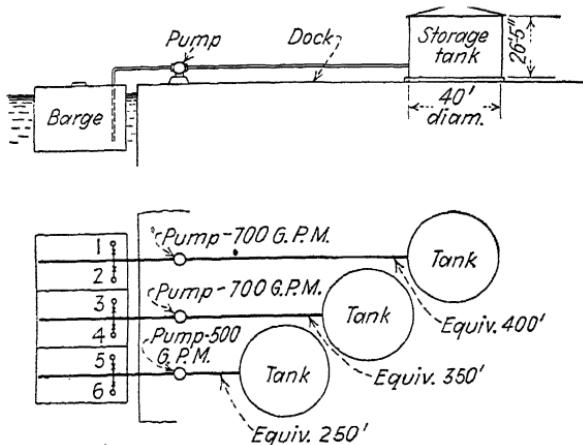


FIG. 148.—Layout for Problem 3.

sq. in. Compute the required pipe size in each of the three cases, and estimate the required theoretical motor capacity for the three pumps, the oil data being as shown below.

OIL DATA

Grade	A.P.I. degrees	Saybolt universal sec.	Specific gravity	Centipoises
6	8	2000	1.01	444.4
5	16	400	0.96	85
4	22	125	0.92	24.4
3	29	60	0.88	9.1
2	34	50	0.86	6.5
1	40	50	0.83	6.3

Solution.—In each of the three cases, it will be sufficient to calculate on the basis of the heavier oil. It is best to work out moduli for these three grades of oil for various pipe sizes, together with the corresponding friction factors as shown in the table at the top of page 253.

The lengths of piping in the three cases are: for No. 6 oil, 250 ft.; for No. 4 oil, 350 ft.; and for No. 2 oil, 400 ft. It is now possible to substitute in the general formula for pressure drop [Eq. (6)] to see how much loss will occur in each case for the various pipe sizes indicated. Since the maximum allowable pump pressure is 75 lb. per sq. in. and since it will require about

Grade of oil	G.p.m.	Pipe size, in.	Pipe diameter, in.	Velocity, ft. per sec.	Modulus, DV _s /Z	Friction factor <i>f</i>
6	500	4	4.03	12.45	0.1141	0.0181
6	500	5	5.05	8.0	0.091	0.023
4	700	3	3.07	30.0	3.475	0.007
4	700	4	4.03	17.4	2.65	0.0078
4	700	5	5.05	11.2	2.13	0.008
2	700	3	3.07	30.0	12.2	0.0052
2	700	4	4.03	17.4	9.28	0.0053

11.5 lb. per sq. in. to overcome the head in a nearly full tank, it follows that only 63.5 lb. per sq. in. can be expended in pumping through the pipes. The figures below show the calculated pressure drop, and also the resultant theoretical motor horsepower required [from Eq. (10) without the efficiencies of motor and pump].

Grade of oil	Pipe size, in.	Pressure drop, lb. per sq. in.	Theoretical motor hp.
6	4	57.5	23.23
6	5	24.1	12.37
4	3	214	94.67
4	4	61.5	31.16
4	5	20.75	15
2	3	170	71
2	4	44.5	22.64

In the case of No. 6 oil, the pressure drop of 57.5 on a 4-in. pipe is fairly close to the limit, and since the loss will increase as the pipe becomes fouled and pitted from use, it is safer to choose a 5 in., especially as the pressure loss is reduced to less than half by so doing, and the theoretical motor load is also reduced substantially (from 23.23 to 12.37). The same reasoning would dictate the choice of a 5-in. pipe for No. 4 oil. On No. 2 oil, it is clear that a 4-in. pipe will be ample from the viewpoint of allowable pressure drop, although the motor size is fairly high. Here it would be a matter of carefully weighing the additional investment in a larger pipe line as against the saving in power cost for operating a smaller motor.

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CHAPTER XIII

OIL PIPING, STRAINERS AND FLOW REGULATORS

Most practical men, whatever may have been their previous occupation or endeavors prior to their becoming associated with the oil-burner industry, will know something about pipework. It is not the intention of this book to present a treatise on practical piping. Plumbing, steam fitting and pipe fitting are such old trades that knowledge of them and information about them are widespread, and almost anyone, from the ubiquitous "handy-man" up, can "cut and thread pipe." Too often, alas, such

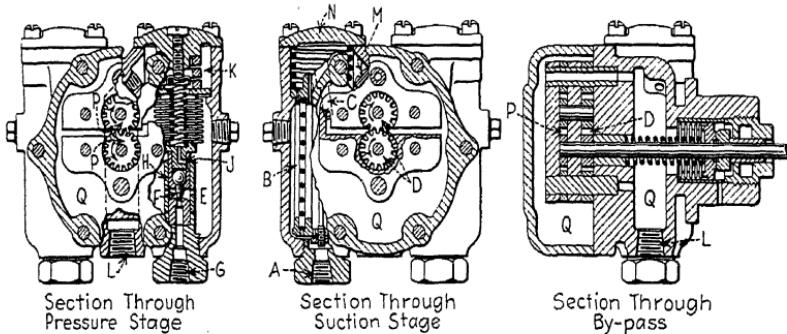


FIG. 149.—Three views of Webster two-stage domestic high-pressure pumping unit.

ability is the sole basis of a mechanic's claim to the status of an installation or service hand. At any rate, it will be sufficient here to discuss some of the high points of this old and honorable profession of pipe fitting as applied to oil-burner work.

Possibly the point of paramount importance is that the pipe be large enough for its job. The older installations of oil burners fairly reek with wholly inadequate pipe sizes, showing total unfamiliarity with hydraulic concepts on the part of the men who installed them, or else gross negligence and carelessness. Such absurdities as tanks with 2-in. fill lines and $\frac{3}{4}$ -in. vent lines are not uncommon. Possibly the perpetrators tried to justify such practice by reasoning that since air is an easily compressible gas, a $\frac{3}{4}$ -in. pipe could readily pass the air displaced by the oil

being pumped into the tank through a 2-in. line. They overlooked the fact that not infrequently a careless oil-truck driver might overflow the tank, and that at such times the $\frac{3}{4}$ -in. line would have to carry out the surplus oil. Besides, since the internal diameter of a 2-in. pipe is 250 per cent that of a $\frac{3}{4}$ in., the volume passed at the same velocity is over six times as great in the larger pipe, so that for equal velocity the smaller pipe would have to handle air compressed to one-sixth the volume at atmospheric pressure. A simple application of Boyle's law shows that the pressure of air under these conditions would be 6 atm., or about 88 lb. per sq. in. And this for a tank designed for a working pressure of 5 lb. per sq. in.! No wonder that the National Board of Fire Underwriters laid down strict and stringent rules for oil-burner installations! Another irritating feature of small pipes arises when modernization of these old oil-burning jobs is attempted. Frequently an old burner operating on light oil is to be replaced by a burner designed for heavy oil. But when it is discovered that the fuel tank is buried under a concrete floor and that the suction line is 50 ft. long or more and of only $\frac{1}{2}$ -in. pipe, whereas the new burner requires a suction line of at least 1 in., and possibly $1\frac{1}{2}$ - or 2-in. pipe, there is an expensive and unpleasant task facing the dealer and his men, which delays and sometimes even defeats the sale by reason of the additional cost involved or because of the impracticability of ripping up a concrete or other finished floor. Let the lessons of past years, costly as they were, serve to emphasize the vital importance of adequate pipe sizes.

The second precept of good workmanship should be that a pipe line between two points should be as short and direct as possible, with as few turns and fittings as the conditions of installation permit. Hydraulicians tell us that every elbow in a pipe introduces an amount of friction equal to 30 diameters, and that 1 tee is equal to 60 diameters of pipe loss. Thus, for a 1-in. pipe, each ell equals about $3\frac{1}{2}$ ft. of additional pipe length, and each tee, 7 ft. The wise old-timer who had observed the effects of fittings on the flow of liquids through pipes used to remark "a dozen ells equals one plug."

Thirdly, make up the joints tightly, and use clean, sound pipe, with the ends reamed out after cutting. And let the ideal of good workmanship carry into the aesthetic too, for a job can be made pleasing in appearance with little or no more effort than required:

to just assemble pipe and fittings into a grotesque nightmare. Valves should be so placed that they are readily accessible, and turned in the direction most suited to using them, *i.e.*, without the necessity of squeezing fingers between pipe barriers in order to reach the valve handle. A valve should be placed near a strainer if there is likelihood of oil flowing out when the strainer is opened. The same applies to check valves, to hold prime on a pump when cleaning the strainer or to prevent backflow of oil from a return line if the pressure regulating valve has to be taken apart. In other words, common sense and pride of workmanship are just as important as connecting the right pipe to the right opening. And a skilled mechanic does not have to be told that sharp dies mean good threads, but that dull or worn dies mean trouble.

Of the thousands of men who work in the pipe trades, but few indeed are familiar with the underlying hydraulic laws which govern the flow of liquids. The preceding chapter gives a brief review of the hydraulics of oil flow, but that discussion has had to presume some knowledge of physics and algebra on the part of the reader. It is difficult indeed to simplify the technology of liquid flow so that it will be comprehensible to those totally untrained in those two subjects, and the general practice is for those who seek such information without having the ability to work it out from fundamental theory and established data to attempt to glean the required facts from handbooks and charts. The author here enters a plea, for the sake of all parties concerned, that those who must make important decisions on size, nature and adaptation of mechanical equipment and who are not, at least to some extent, versed in the selection of such equipment by virtue of academic training or long experience, should consult with someone qualified to pass competent opinion upon any selection made.

PIPE DATA

For many years the standard conduit for liquids in ordinary use has been so-called iron pipe, which of course in reality is steel pipe. To distinguish regular steel pipe from special wrought-iron pipe, it is customary to prefix the word genuine, referring to it as genuine wrought-iron pipe. Both steel and wrought-iron pipe are available in two strengths, designated as standard and as

TABLE XXVIII.—STANDARD STEEL PIPE—BLACK AND GALVANIZED*
All Weights and Dimensions Are Approximate

Nominal diameter, in.	Diameters		Thickness, in.	Weight per ft.		Threads per in.	Couplings		
	External, in.	Internal, in.		Plain ends, lb.	Threads and couplings, lb.		Diameter, in.	Length, in.	Weight, lb.
3/8	0.405	0.269	0.068	0.244	0.245	27	0.562	7/8	0.030
1/4	0.540	0.364	0.088	0.424	0.425	18	0.685	1	0.044
5/16	0.675	0.493	0.091	0.567	0.568	18	0.848	1 1/8	0.072
3/16	0.840	0.622	0.109	0.850	0.852	14	1.024	1 1/8	0.118
7/16	1.050	0.824	0.113	1.130	1.134	14	1.281	1 1/8	0.214
1	1.315	1.049	0.133	1.678	1.684	11 1/2	1.576	1 1/8	0.350
1 1/4	1.660	1.380	0.140	2.272	2.281	11 1/2	1.950	2 1/8	0.546
1 1/2	1.900	1.610	0.145	2.717	2.731	11 1/2	2.218	2 1/8	0.758
2	2.375	2.067	0.154	3.652	3.678	11 1/2	2.760	2 1/8	1.233
2 1/2	2.875	2.469	0.203	5.793	5.819	8	3.276	2 1/8	1.755
3	3.500	3.068	0.216	7.575	7.616	8	3.948	3 1/8	2.549
3 1/2	4.000	3.548	0.226	9.109	9.202	8	4.591	3 1/8	4.328
4	4.500	4.026	0.237	10.790	10.889	8	5.091	3 1/8	5.410
4 1/2	5.000	4.506	0.247	12.538	12.642	8	5.591	3 1/8	5.984
5	5.563	5.047	0.258	14.617	14.810	8	6.296	4 1/8	9.158
6	6.625	6.065	0.280	18.974	19.185	8	7.358	4 1/8	10.823
7	7.625	7.023	0.301	23.544	23.769	8	8.358	4 1/8	12.390
8	8.625	8.071	0.277	24.696	25.000	8	9.420	4 5/8	15.843
8	8.625	7.981	0.322	28.554	28.809	8	9.420	4 5/8	15.843
9	9.625	8.941	0.342	33.907	34.188	8	10.420	5 1/8	19.752
10	10.750	10.192	0.279	31.201	32.000	8	11.721	6 1/8	33.923
10	10.750	10.136	0.307	34.240	35.000	8	11.721	6 1/8	33.923
10	10.750	10.020	0.365	40.483	41.132	8	11.721	6 1/8	33.923
11	11.750	11.000	0.375	45.557	46.247	8	12.721	6 1/8	36.970
12	12.750	12.090	0.330	43.773	45.000	8	13.958	6 1/8	48.266
12	12.750	12.000	0.375	49.562	50.706	8	13.958	6 1/8	48.266

* Courtesy of National Tube Company.

† These sizes are listed as standard but are not commonly used.

The permissible variation in weight is 5 per cent above and 5 per cent below.

Furnished with threads and couplings and in random lengths unless otherwise ordered.

Taper of threads is 3/4-in. diameter per foot length for all sizes.

The weight per foot of pipe with threads and couplings is based on a length of 20 ft., including the coupling, but shipping lengths of small sizes will usually average less than 20 ft. All weights given in pounds. All dimensions given in inches.

extra strong (as well as special higher strengths), and both standard and extra strong kinds of pipe, respectively, carry identical dimensions for either steel or wrought. Full data on both are

TABLE XXIX.- CIRCUMFERENCES, TRANSVERSE AREAS AND SURFACES FOR STANDARD STEEL PIPE AND LINE PIPE*

Nominal diameter, in.	Diameter, in.		Thickness, in.	Circumference, in.			Transverse area, sq. in.			Length of pipe per sq. ft., ft.		Length of pipe containing 1 cu. ft., ft.
	External	Internal		External	Internal	External	Internal	Metal	External surface	Internal surface		
3/8	0.405	0.269	0.068	1.272	0.845	0.129	0.057	0.072	9.431	14.199	1533.775	
1/4	0.540	0.364	0.088	1.696	1.144	0.229	0.104	0.125	7.073	10.493	1883.789	
3/16	0.675	0.493	0.091	2.121	1.549	0.358	0.191	0.167	5.658	7.748	754.360	
1/8	0.840	0.622	0.109	2.639	1.954	0.554	0.304	0.250	4.547	6.141	473.906	
5/32	1.050	0.824	0.113	3.299	2.589	0.866	0.533	0.333	3.637	4.635	270.034	
1	1.315	1.049	0.133	4.131	3.296	1.358	0.864	0.494	2.904	3.641	166.618	
1 1/4	1.660	1.380	0.140	5.215	4.335	2.164	1.495	0.669	2.301	2.768	96.275	
1 1/2	1.900	1.610	0.145	5.969	5.058	2.835	2.036	0.799	2.010	2.372	70.733	
2	2.375	2.067	0.154	7.461	6.494	4.430	3.355	1.075	1.608	1.847	42.913	
2 1/2	2.875	2.469	0.203	9.032	7.757	6.492	4.788	1.704	1.328	1.547	30.077	
3	3.500	3.068	0.216	10.996	9.638	9.621	7.393	2.228	1.091	1.245	19.479	
3 1/2	4.000	3.548	0.226	12.566	11.146	12.566	9.886	2.680	0.954	1.076	14.565	
4	4.500	4.026	0.237	14.137	12.648	15.904	12.730	3.174	0.848	0.948	11.312	
4 1/2	5.000	4.506	0.247	15.708	14.156	19.635	15.947	3.888	0.763	0.847	9.030	
5	5.563	5.047	0.258	17.477	15.856	24.306	20.006	4.300	0.686	0.756	7.198	
6	6.625	6.065	0.280	20.813	19.054	34.472	28.891	5.581	0.576	0.629	4.984	
7	7.625	7.023	0.301	23.955	22.063	45.664	38.738	6.926	0.500	0.543	3.717	
8	8.625	8.071	0.277	27.096	25.356	58.426	51.161	7.265	0.443	0.473	2.815	
8	8.625	7.981	0.322	27.096	25.073	58.426	50.027	8.399	0.443	0.478	2.878	
9	9.625	8.941	0.342	30.238	28.089	72.760	62.786	9.974	0.396	0.427	2.294	
10	10.750	10.192	0.279	33.772	32.019	90.763	81.585	9.178	0.355	0.374	1.765	
10	10.750	10.136	0.307	33.772	31.843	90.763	80.691	10.072	0.355	0.376	1.785	
10	10.750	10.020	0.365	33.772	31.479	90.763	78.855	11.908	0.355	0.381	1.826	
11	11.750	11.000	0.375	36.914	34.558	108.434	95.033	13.401	0.325	0.347	1.515	
12	12.750	12.090	0.330	40.055	37.982	127.676	114.800	12.876	0.299	0.315	1.254	
12	12.750	12.000	0.375	40.055	37.699	127.676	113.097	14.579	0.299	0.318	1.273	
14 O.D.	14.000	13.250	0.375	43.982	41.626	153.938	137.886	16.052	0.272	0.288	1.044	
15 O.D.	15.000	14.250	0.375	47.124	44.768	176.718	159.485	17.230	0.254	0.268	0.903	
16 O.D.	16.000	15.250	0.375	50.265	47.909	201.062	182.654	18.408	0.238	0.250	0.788	
17 O.D.	17.000	16.214	0.393	53.407	50.938	226.980	206.476	20.504	0.224	0.235	0.697	
18 O.D.	18.000	17.182	0.409	56.549	53.979	254.469	231.866	22.603	0.212	0.222	0.621	
20 O.D.	20.000	19.182	0.409	62.832	60.262	314.159	288.986	25.173	0.191	0.199	0.498	

* Courtesy of National Tube Company.

given in Tables XXVIII to XXXI. Within the past few years, copper tubing has become very popular in oil-burner work, especially for domestic installations. Copper tubing offers the distinct advantages of labor saving and elimination of pipe

TABLE XXX.—EXTRA STRONG STEEL PIPE—BLACK AND GALVANIZED*
All Weights and Dimensions Are Approximate

Nominal diameter, in.	Diameter, in.		Thickness, in.	Weight per ft. plain ends, lb.†
	External	Internal		
1/8	0.405	0.215	0.095	0.314
1/4	0.540	0.302	0.119	0.535
3/8	0.675	0.423	0.126	0.738
1/2	0.840	0.546	0.147	1.087
5/8	1.050	0.742	0.154	1.473
1	1.315	0.957	0.179	2.171
1 1/4	1.660	1.278	0.191	2.996
1 1/2	1.900	1.500	0.200	3.631
2	2.375	1.939	0.218	5.022
2 1/2	2.875	2.323	0.276	7.661
3	3.500	2.900	0.300	10.252
3 1/2	4.000	3.364	0.318	12.505
4	4.500	3.826	0.337	14.983
‡ 4 1/2	5.000	4.290	0.355	17.611
5	5.563	4.813	0.375	20.778
6	6.625	5.761	0.432	28.573
‡ 7	7.625	6.625	0.500	38.048
8	8.625	7.625	0.500	43.388
‡ 9	9.625	8.625	0.500	48.728
10	10.750	9.750	0.500	54.735
‡ 11	11.750	10.750	0.500	60.075
12	12.750	11.750	0.500	65.415

* Furnished with plain ends and in random lengths unless otherwise ordered.

† The permissible variation in weight is 5 per cent above and 5 per cent below.

‡ These sizes listed as standard but not commonly used.

joints, with their always attendant possibilities of leaks. Data on copper tubing are shown in Table XXXII.

STRAINERS FOR FUEL OIL

Fuel-oil strainers can be classified as perforated metal, wire mesh, metal disk and filter cloth. Roughly, their applications vary in the order named from the heaviest oils to the lightest.

TABLE XXXI.—CIRCUMFERENCES, TRANSVERSE AREAS AND SURFACES FOR EXTRA-STRONG STEEL PIPE*

Nominal diameter, in.	Diameter, in.		Thickness, in.	Circumference, in.		Transverse area, sq. in.			Length of pipe per sq. ft., ft.		Length of pipe containing 1 cu. ft., ft.
	External	Internal		External	Internal	External	Internal	Metal	External surface	Internal surface	
1/8	0.405	0.215	0.095	1.272	0.675	0.129	0.036	0.093	9.431	17.766	3966.393
1/4	0.540	0.302	0.119	1.696	0.949	0.229	0.072	0.157	7.073	12.648	2010.290
3/8	0.675	0.423	0.126	2.121	1.329	0.358	0.141	0.217	5.658	9.030	1024.689
1/2	0.840	0.546	0.147	2.639	1.715	0.554	0.234	0.320	4.547	6.995	615.017
5/8	1.050	0.742	0.154	3.299	2.331	0.866	0.433	0.433	3.637	5.147	333.016
1	1.315	0.957	0.179	4.131	3.007	1.358	0.719	0.639	2.904	3.991	200.193
1 1/4	1.660	1.278	0.191	5.215	4.015	2.164	1.283	0.881	2.301	2.988	112.256
1 1/2	1.900	1.500	0.200	5.969	4.712	2.835	1.767	1.068	2.010	2.546	81.487
2	2.375	1.939	0.218	7.461	6.092	4.430	2.953	1.477	1.608	1.969	48.766
2 1/2	2.875	2.323	0.276	9.032	7.298	6.492	4.238	2.254	1.328	1.644	33.976
3	3.500	2.900	0.300	10.996	9.111	9.621	6.605	3.016	1.091	1.317	21.801
3 1/2	4.000	3.364	0.318	12.566	10.568	12.566	8.888	3.678	0.954	1.135	16.202
4	4.500	3.826	0.337	14.137	12.020	15.904	11.497	4.407	0.848	0.998	12.525
4 1/2	5.000	4.290	0.355	15.708	13.477	19.635	14.455	5.180	0.763	0.890	9.962
5	5.563	4.813	0.375	17.477	15.120	24.306	18.194	6.112	0.686	0.793	7.915
6	6.625	5.761	0.432	20.813	18.099	34.472	26.067	8.405	0.576	0.663	5.524
7	7.625	6.625	0.500	23.955	20.813	45.664	34.472	11.192	0.500	0.576	4.177
8	8.625	7.625	0.500	27.096	23.955	58.426	45.663	12.763	0.443	0.500	3.154
9	9.625	8.625	0.500	30.238	27.096	72.760	58.426	14.334	0.396	0.442	2.465
10	10.750	9.750	0.500	33.772	30.631	90.763	74.662	16.101	0.355	0.391	1.929
11	11.750	10.750	0.500	36.914	33.772	108.434	90.763	17.671	0.325	0.355	1.587
12	12.750	11.750	0.500	40.055	36.914	127.676	108.434	19.242	0.299	0.325	1.328

* Courtesy of National Tube Company.

TABLE XXXII.—DATA ON COPPER TUBING
(UNDERWRITERS' SPECIFICATIONS)

Equivalent iron-pipe-size, in.	Outside diameter, in.	Inside diameter, in.	Wall thickness, in.
...	3/8 or less	0.049
5/8	0.500	0.402	0.049
1/2	0.625	0.527	0.049
3/4	0.875	0.745	0.065
1	1.125	0.995	0.065

Perforated metal cylinders or cones, closed at one end and open at the other, are commonly used in bulk operations where the chief purpose of the strainers is simply to protect pumping equip-

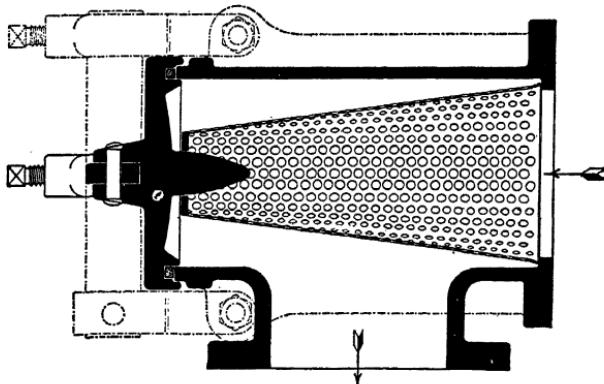


FIG. 150.—Schutte and Koerting perforated metal strainer.

ment by keeping out of the pumps relatively large foreign bodies—pebbles, sticks, etc. The cylinders are contained within castings which are so formed that the flow of oil through the assembly is

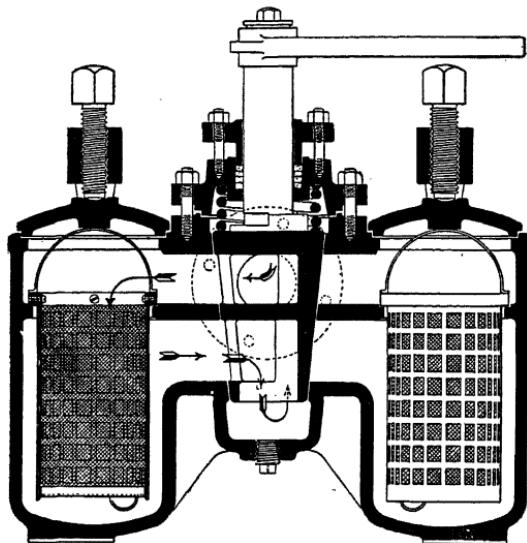


FIG. 151.—Schutte and Koerting duplex wire-mesh strainers.

directed into the open top of the cylinder and out through the walls, with provision made against short circuiting—*i.e.*, of oil passing from inlet of strainer body to outlet without having to

pass through the cylinder. Thus the solid matter which the strainer removes from the liquid is caught and retained within the inside of the cylinder, so that by lifting the cylinder out of the strainer body all the solid matter is removed with it. The cylinder or cone (Fig. 150) may then be cleaned and replaced, and the strainer body capped up for another period of use.

Perforated metal cylinders are sometimes applied to smaller units also, such as moderate-sized industrial oil burners using the heavier grades of fuel oil. The perforations are then smaller in diameter and more numerous, in order that the smaller sized particles of sediment will be retained without there being too much resistance to flow through the strainer. Usually a strainer of this type will be followed by another strainer of wire mesh to perform the final cleaning operation just before the fuel enters the nozzle or atomizer.

Wire-mesh strainers are made in a variety of shapes and sizes, and constitute the most important group. The mesh itself is a metallic cloth woven from wire, and a wide range of variation is possible in the diameter of the wire and the spacing between them. Some meshes are woven of such fine wire placed so closely together that only the very lightest fuels, such as kerosene or distillate, will pass through them. Commercial wire-mesh strainers range from 10 wires per linear inch to 100 wires, and are designated by the number of wires per inch, as 30 mesh or 100

TABLE XXXIII.—DATA ON WIRE MESH FOR OIL STRAINERS
Either Bronze- or Monel-wire Cloth

Mesh	Diameter of wire, in.	No. of wire on Washburn and Moen gauge	Width of opening, in.	Proportion of open space to total screen area, per cent
10	0.015	29	0.085	72.3
20	0.0135		0.0365	53.3
30	0.012		0.0213	40.8
40	0.0104	34	0.0146	34.1
50	0.0085	37	0.0115	33.1
60	0.0075	39	0.0092	30.5
70	0.0065		0.0078	29.8
80	0.0055	45	0.007	31.4
90	0.005	47	0.005	25.0
100	0.0045		0.0055	30.3

mesh, etc. To be completely descriptive, however, the terminology should also state the wire diameter.

Formula for open area in any mesh is given by

$$\text{Open area} = (1 - ND) \times (1 - nd)$$

in which N = number of wires in the warp.

n = number of wires in the shoot.

D = diameter of wire in the warp.

d = diameter of wire in the shoot.

For 100-mesh cloth in above table, the open area is 30.2 per cent.

As in the case of the perforated metal cylinder, the baskets for these strainers should be permitted to catch sediment on the inside, to facilitate removal from the strainer body. Some strainer bodies are constructed so that oil passes from the outside to the inside of the basket, so that sediment is retained on the outer surface of the basket. Subsequently, as the basket is being removed from the body, portions of the accumulated sediment are washed back into the strainer body and must be drained out by removing a plug at the bottom of the body. In other forms, the body cap is at the bottom, so that when opened the basket and all other contents of the body are dropped out. The objection to this is the difficulty of replacing the basket and strainer cap after each cleaning operation.

The third group of strainers is built up of a series of metallic disks, stacked in a column with close clearances between consecutive disks. The disks are arranged to communicate into a common tube at their centers. Oil enters into the clearances between the disks at their circumferences; particles of solid matter of greater dimensions than the clearance are stopped at the outer surface of the column of disks and from there are removed by scraping the surface of the column. This cleaning operation is



FIG. 152.—Wire-mesh strainers for domestic burners. (*Courtesy of Monarch Manufacturing Works, Inc.*)

accomplished by turning the crank of a shaft which extends through the strainer body to a convenient point on the top. The shaft communicates to a scraping edge which is rotated over the

column surface by turning the crank outside. The sediment thus scraped clear falls to the bottom of the strainer body and can be released at intervals by removing the bottom plug. Figure 153 gives a sectional view.

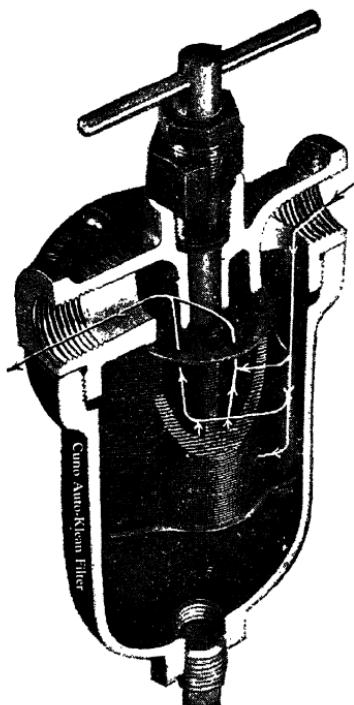


FIG. 153.—Disk oil strainer. (*Courtesy of Cuno Engineering Company.*)

The filter-cloth strainer is an adaptation of the type used on automobiles for filtering lubricating oil. In order that as much filtering surface as possible may be contained within a strainer body, the cloth is arranged in layers folded into an accordion-like form over a wire skeleton. Obviously, it is imperative that the flow of oil be from the outside toward the inside, for if the reverse occurred, cleaning of the filter surface would be impossible.

In practice, filter-cloth strainers are restricted to the lighter or domestic fuels, as they foul too rapidly when used on heavier industrial oils.

In general, strainers, regardless of type, must be designed for minimum loss of head as the fluid passes through. The initial pressure drop through the strainer becomes progressively greater as the accumulation of sediment on the cleaning surface impedes flow. The frequency of cleaning is therefore determined by the total amount of original surface, the percentage of sediment in the oil, and the permissible pressure drop through the strainer. Wire-mesh strainers are frequently used on suction lines, where, of course, the force creating flow is the atmospheric pressure on the storage tank. A vacuum gauge will therefore show the amount of vacuum needed to maintain flow through the strainer; when the vacuum-gauge reading has mounted to a point approach-

ing or equalling 28 in., flow will cease. In practice, strainers should be cleaned before this limiting pressure drop has been reached. When the strainer is located in a pressure line, then pressure gauges on the inlet and outlet sides will show the pressure drop through it, so that cleaning may be timed from a knowledge of the pressures available in the pumping system.

OIL-PRESSURE REGULATION

The almost universal method of regulating oil pressure on oil-burning systems is to discharge the pump against some type of relief valve placed near the pump outlet. These relief valves vary considerable in their construction, but all of them contain a balancing device to create a pressure against the pump discharge, and this device is usually adjustable to give some range of pressure variation.

In the simplest type (used frequently where operating pressures are between 5 and 25 lb. per sq. in., and occasionally for higher pressures) a valve stem and plunger are forced against a valve seat by an adjustable spring. The spring tension determines the pressure which the pump must develop to push oil past the plunger. Hence oil taken to the nozzle from any point between the pump outlet and the relief valve will be delivered at the pressure so determined. Details of a typical pressure-relief valve appear in Fig. 154. This type of pressure regulator has the virtue of simplicity, although there are several objectionable features.

Both the valve seat and the plunger are subject to eroding and corroding action, and wire drawing also exerts its effect upon these parts. As a result, the plunger becomes scored and the seat grooved after a period of use, so that pressure regulation first becomes irregular and then may become impossible. To remedy this condition the plunger and seat must be resurfaced by grinding with abrasive, much the same as with the valves of an automobile. Another objection is the tendency of the spring gradually to lose its elasticity, ultimately requiring its replacement. Most serious is the fact that this type of valve is easily clogged or rendered inoperative by the entrance into it of small solid particles or

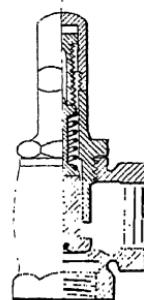


FIG. 154.—
Spring-operated
pressure-relief or
regulating
valve.

stringy materials such as tars and greases. For handling heavy fuel oil, therefore, this is not an ideal means of pressure regulation.

Domestic-type pressure-atomizing burners invariably employ some form of a compound pressure-relief valve in conjunction

with a pressure cutout, the function of the latter being to stop the flow of oil to the nozzle at the moment that the pressure drops below some predetermined minimum. These two parts, the pressure regulator and the pressure cutoff, formerly were housed separately, but it is almost universal practice today to place the two together as a single piece of equipment, with adjustments provided for at least varying the pressure relief and sometimes for varying the differential between relief pressure and cutout pressure. Commercial forms of this type have been made in three classes: piston, bellows and diaphragm operated. The piston type predominates in current practice.

In the piston-type pressure-regulating and cutoff valve, the by-pass for the surplus oil may be either in the cylinder wall or in the piston itself. When placed in the

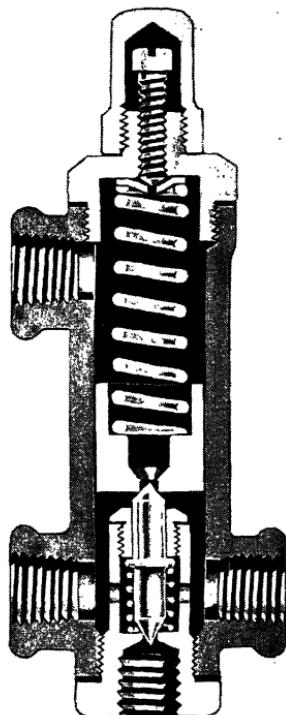


FIG. 155.—Piston-type pressure-regulating and cutoff valve, domestic pressure-atomizing burners
(Courtesy of Monarch Manufacturing Works, Inc.)

pump exceeds the force required to compress the spring in the valve to such an extent that the piston moves sufficiently to uncover the by-pass port. When the by-pass port is in the piston head, the oil pressure is required to compress the spring until the motion of a needle on the valve stem uncovers the port.

Cutoff of oil to the nozzle is accomplished by a needle valve attached to the piston, or used in conjunction with the piston, the needle receiving its closing pressure from a spring as the pump

by the piston until the developed pressure on the oil leaving the

pressure diminishes. The customary differential between maintained pressure during pump operation and the closing pressure of cutoff to nozzle is about 30 lb. when the maintained pressure is 75 lb. (the differential decreasing as the developed or maintained pressure increases) to as low as 20 lb. differential at 150 lb. pump pressure. Standard valves are constructed to operate at pump pressures between 75 and 125 lb., although special valves are constructed for operation as high as 400 lb. pressure.

These piston-type pressure-regulating valves are all of the so-called leakage type—meaning that oil will leak past the piston, it being impossible to make the piston fit in the cylinder so tight as to avoid it. Even on a new valve, the leakage will be so high that the nozzle delivery cannot exceed 50 per cent of the pump capacity. If nozzles of higher capacity are used, the pump and valve will not maintain operating pressure. As the valve wears, the leakage increases and capacity is correspondingly reduced. Frequently, lack of oil pressure may be due to valve wear rather than to pump wear; although if pressure cannot be obtained with a new valve or valve known to be in good working condition, then the trouble must be sought in the pump.

Springs for these valves are made of stainless steel, monel metal and cadmium-plated steel or bronze. Conventional materials are brass or bronze for the cylinder, and stainless steel for the piston. All parts of these valves must be accurately made and held within close dimensions.

These regulators are subject to the same objections as the pressure-relief valve first described, but in the domestic valves seat erosion and loss of spring tension are more marked because of the higher operating pressures, while clogging is of little concern since domestic fuels are lighter and cleaner and hence not likely to carry sediment.

Certain manufacturers of fuel-oil equipment employ unusual means of controlling oil pressure, such as variation of eccentricity of pump gears or variation of gear mesh by lateral sliding of the gears, but to date none of these has come into extensive use. No doubt new mechanisms for pressure regulation will be developed.

METERING, OR FEED REGULATION

The final operation in a fuel transmission system is that of metering, or regulating the feed to the nozzle or atomizer. There

are two distinct phases to this operation: the first, which is the one considered in this chapter, being the mechanism of hydraulic regulation of the flow; the second being the method of actuating this mechanism and hence of proportioning the fuel flow in accordance with load or process demands. Full treatment is given thereon in Chaps. VIII and XV.

The first and still the commonest means employed for regulating oil flow is the needle valve, both in the angle and straight-through forms; for light oils it is a fairly satisfactory valve. Many forms are made with indicating devices to show the amount of rotation from the closed position. For heavier oils, however, they exhibit a tendency toward flow restrictions, probably because of film formation. In the needle valve, the space through which the fluid flows is distributed as a ring between the two concentric circles of the valve stem and the valve seat. This ring very often is so narrow that the viscous oils tend to cling to both sides, and if the surface tension is sufficiently strong, the resultant film which is gradually stretched across the opening slowly diminishes the flow until it stops completely. Operators often try to overcome this by raising the oil pressure at the pump, but that is exactly the wrong thing to do. Higher pressure merely increases the speed of flow, so that the annular opening of the needle valve must be even smaller for the same quantity, and the tendency toward film formation increases. To correct diminishing flow in a needle valve, drop the pressure to the lowest possible point so that the valve may be opened as wide as possible for the permissible quantity passed. This will retard and sometimes overcome the tendency toward film formation. Sometimes the same diminution of flow occurs because of colloidal sediment which cannot be removed at the strainers; the same rule holds—running at the lowest possible pressure to prevent the sediment from depositing on the valve seat, which is the point of greatest restriction after the strainers.

Globe valves are seldom used in oil systems, both because of the high-pressure drop which they induce in the line and because the slight clearance between the disk and seat gives rise to the same film formation or sediment deposit as with needle valves. Gate valves are, of course, hardly adaptable to control but are commonly used as shutoff valves.

A popular adaptation of the needle valve is the metering, or V-notch valve, in which the stem is a cylinder fitting closely through the seat opening. When the stem (Fig. 156) is turned completely down, the upper end of the cylinder, which broadens out into a cone, brings the seat surface into tight contact with this cone and flow ceases. Cut longitudinally into the surface of the cylinder is a V notch which has a varying cross section. Turning the valve stem therefore exposes a varying cross section of the V for flow of liquid, so that close control can be had. The same amount of opening as would exist in a needle valve for the same flow under identical conditions is therefore condensed from a ring into a triangle so that the smallest dimension of the opening is much greater and the tendency toward film formation or sediment deposition is reduced. This V-notch valve serves well with most industrial fuels except the very heaviest.

Leaving the realm of valves, the next sort of control device is the orifice plate. An orifice offers the advantages that no working parts are needed in the device itself, and that flow through it can be controlled by simply varying the pressure behind it. The flow is independent of the film restrictions because of the full circular opening. Against the orifice as a regulating device are the facts that with any given orifice the flow can be varied only to the maximum and minimum limits set by available pressure variation; beyond these limits, it becomes necessary to use another orifice plate, with a larger or smaller orifice as the case may be. Orifice plates are also occasionally used with standpipes, so that the head acting on the upstream side is varied by raising or lowering the level of liquid in the standpipe. This method is not easily adaptable to industrial use.

On pressure-atomizing oil burners, the orifice in the nozzle tip serves the same functions of determining the rate of flow for any given oil pressure behind it, although the orifice also serves as part of the atomizing system, and the slots in the nozzle tip also influence capacity by retarding the oil passage in proportion to the slot size.

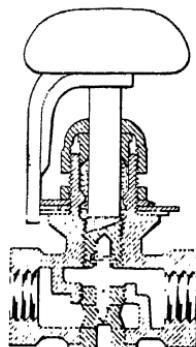


FIG. 156.—V-notch metering valve.

VISCOSITY COMPENSATION

These two methods of control, valves and orifices, embody oil pressure as one of the variables, for regardless of the valve or orifice used, flow is determined by the pressure head existing on the upstream side, or, more properly, by the pressure differential

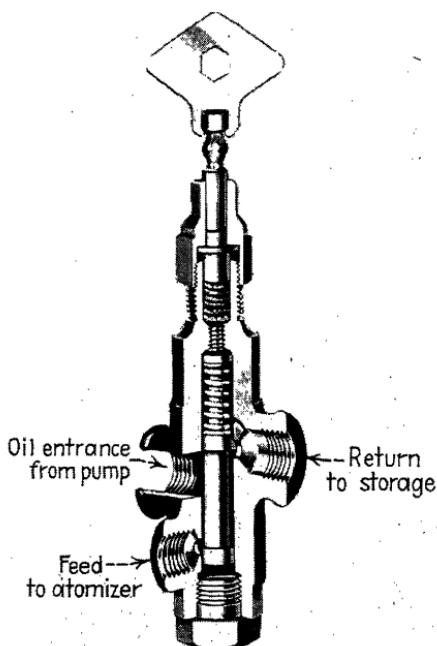


FIG. 157.—Uniflow oil-regulating and viscosity-compensating valve embodying emergency pressure relief also. (*Courtesy of Ace Engineering Company.*)

between the upstream and downstream side, which is a function of the upstream pressure. Since pressure regulators are not as yet perfect in their regulation, it follows that metering devices dependent upon them will reflect a variation in flow equivalent to the variation in pressure. Changes in oil viscosity will also cause changes in delivery through a valve or orifice. Hence a regulator independent of oil pressure will eliminate that variable, and at least one form is available which also eliminates variation due to viscosity changes.

One commercial form of such a flow regulator (Fig. 157) depends for its operation upon the uniform and unvarying delivery of a displacement pump, such as an internal-gear type, regardless of changes in the fuel oil's viscosity, temperature or other physical attributes. This uniform delivery is taken into the central cylindrical bore of the uniflow valve, as it is called. There are two outlets from this central bore, one leading back to the storage tank, the other to the burner. A piston working

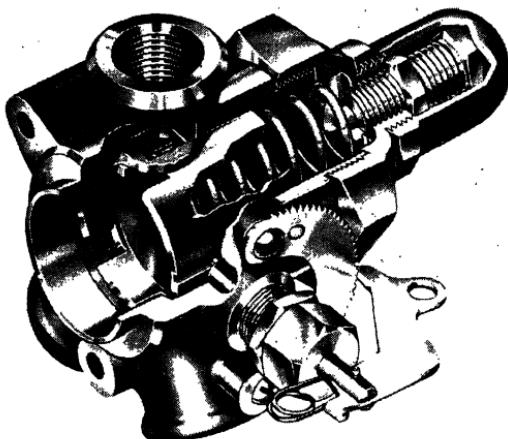


FIG. 158.—Combination regulating, viscosity-compensating and pressure-relief valve. (*Courtesy of S. T. Johnson Company.*)

back and forth in this bore will vary, by means of a screw adjustment, the area exposed by the two outlets of the central chamber. With any position of the piston, therefore, a ratio is determined between the outlet to the tank and the outlet to the burner. The discharge from the pump must therefore divide in conformance with this ratio, and since the discharge is a constant, regardless of the physical state of the oil, the feed to the burner is a constant.

MISCELLANEOUS DEVICES

Some devices utilize the uniform delivery of a gear-type pump to establish the feed itself, by providing a means of varying the mesh of the gears. For any given mesh, the discharge is a constant, and this discharge is used as the burner feed. In such cases the surplus from the main pump, ahead of the metering

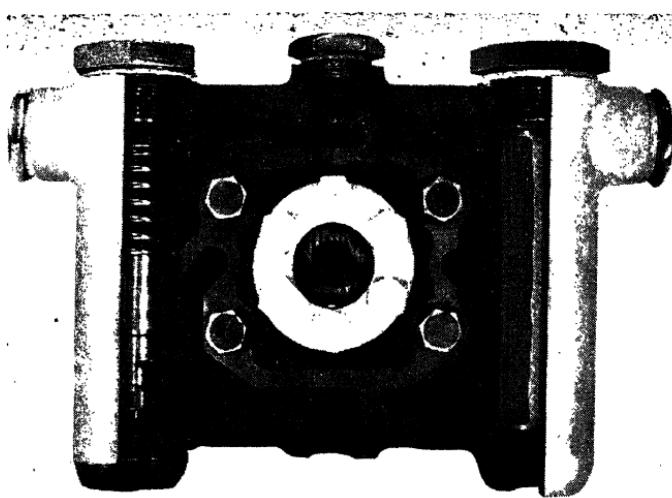


FIG. 159.—Domestic fuel unit for pressure-atomizing burner, comprising pump, strainer, pressure-regulating valve and oil shutoff. (*Courtesy of Viking Pump Company.*)

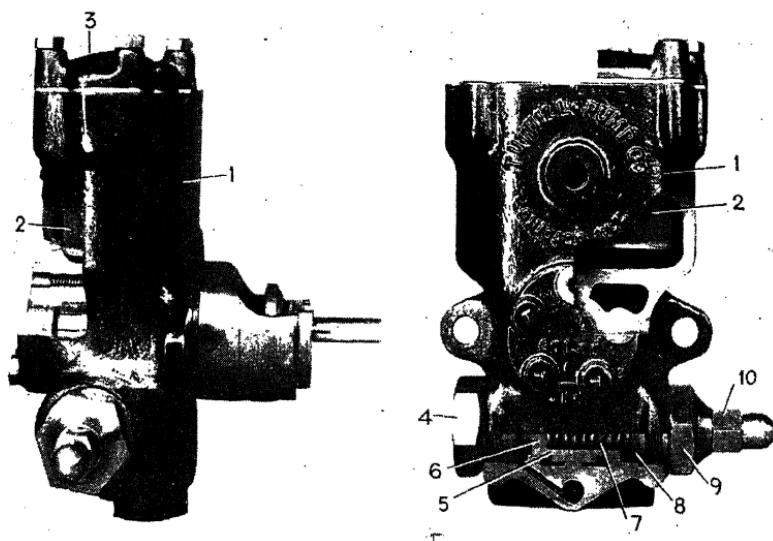


FIG. 160.—Domestic fuel unit for pressure-atomizing burners. 1, housing; 2, strainer; 3, strainer cover; 4, valve cap; 5, valve bushing; 6, valve piston; 7, valve spring; 8, valve-spring cover; 9, valve bonnet; 10, valve adjusting-screw cover. (*Courtesy of Tuthill Pump Company.*)

pump, is returned to storage. This method however implies the use of two pumps, which is not always desirable.

A recent interesting development is a fuel unit for domestic oil-burner use which embraces three or four elements of the pumping system (usually made by at least two different manufacturers) such as pump, strainer, pressure-regulating valve and sometimes a pressure gauge or vacuum gauge. Two typical forms are illustrated in Figs. 159 and 160.

CHAPTER XIV

COMBUSTION CHAMBERS FOR HEATING PLANTS

The skill and knowledge and experience of an oil-burner manufacturer, combined with his factory equipment and personnel, can turn out a satisfactory oil burner. But when this burner is removed from its shipping crate by the dealer, preparatory to its installation on the premises of the purchaser, then another and entirely different group of skill, knowledge, experience, equipment and personnel become responsible for the completion of the work initiated in the oil-burner factory. Now it remains for the dealer, or other party upon whom the installation devolves, to place that burner in operation, and that can be successful only if, and when, all the adjuncts to installation and operation are correctly performed.

Perhaps no attribute or factor affecting this phase of oil-burner practice is quite so important as good common sense. The rule of reason is probably more effective than all other rules that have been formulated or offered in support of one theory or another. By this the author does not mean that so-called rule of thumb, acquired by operating men from their years of experience, is always or even frequently to be accepted without question or to be substituted for sound technical or scientific fact. It is rather a matter of determining how closely theory and practice can be made to conform to one another.

Suppose for instance that a person wholly unfamiliar with oil-burner work were suddenly confronted with the task of selecting an oil strainer for a certain application, and were given as data that the particular oil to be used contained $\frac{3}{4}$ per cent sediment and that the pump capacity was 100 g.p.h., the strainer to be placed in the suction line ahead of the pump inlet. Reasoning that the strainer capacity should at least be sufficient to run for a 10-hr. day without requiring cleaning, he would calculate that the total gallons pumped in 10 hr. would be 1000 gal., weighing about 8000 lb., and that $\frac{3}{4}$ per cent would be 60 lb. If, accord-

ingly, he were to design for a strainer to hold 60 lb. of sediment, the selection would be a strainer so large as to be out of all proportion to what would actually be needed. The indicated process of reasoning overlooks two facts: much of the sediment revealed by a chemical test of the oil is in so fine a state of aggregation that it passes through the strainer meshes, regulating valves and nozzles, and is still in the oil when atomized, and

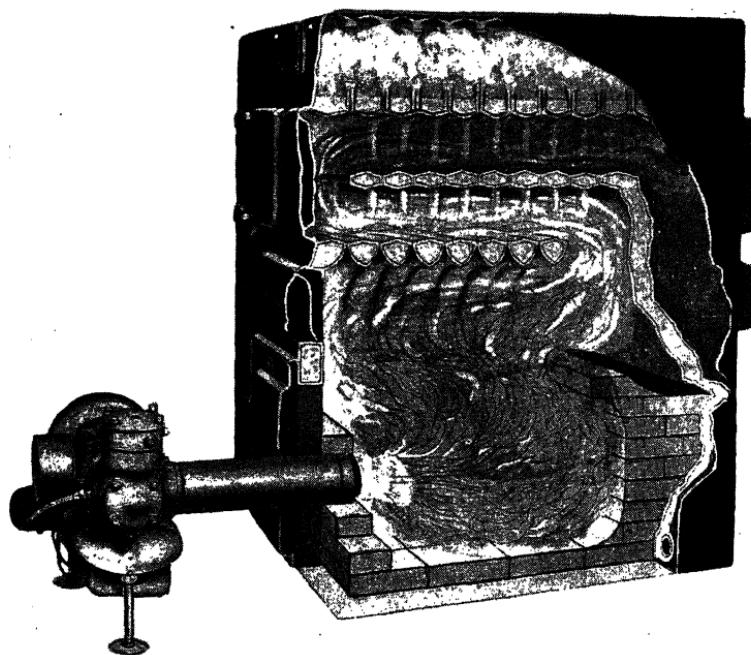


FIG. 161.—General view of firebox and flue passages through sectional heating boiler. (*Courtesy of Williams Oil-O-Matic Heating Corporation.*)

again, of the 100 gal. handled each hour by the pump, all is not burned, the surplus returning to storage, to be circulated repeatedly through the pump and strainer, so that the same oil may be strained several times.

Most manufacturers of oil burners supply fairly comprehensive installation instructions with their equipment. Such instructions should always be followed as closely as local conditions permit. Piping and electric wiring are arts or trades in themselves, and oil burners represent but a small portion of their field of useful-

ness. Hence it is but necessary to study that much which is peculiar to our purposes. But combustion chambers are inherently an almost integral part of the burner itself, and so really must be thoroughly investigated to understand the fundamental concepts as well as the usual applications.

FUNCTIONS OF THE FIREBOX

Primarily, the firebox was originally intended to serve as a refractory chamber within which combustion could occur, the refractory walls and floor serving to protect the boiler setting from injury from the fire. It was soon learned that when the refractory itself attained a temperature of incandescence that its

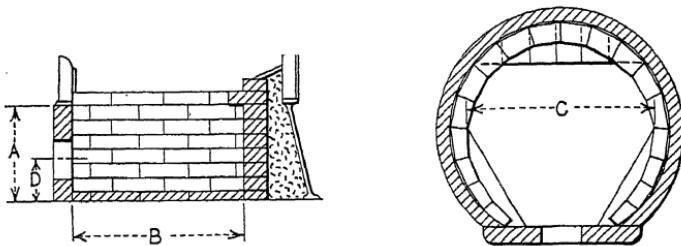


FIG. 162.—Vertical round-boiler firebox for gun-type burners. See Table XXXIV or values of indicated dimensions.

presence exerted a catalytic effect upon the burning fuel and assisted in securing more complete combustion. Finally, it also became apparent that much heat was being radiated from the glowing brick, and that by proper shaping or design of the box, considerable heat could be transferred into the boiler by direct radiation from the refractory.

In the conventional boiler or warm-air furnace, the burner should always be set in a position as low as the setting permits. For all burners outside the boiler or furnace, this means that the burner should be below the firing door—usually in the ashpit. For burners placed inside the boiler, the ashpit should be utilized for the supports which hold the burner in place, so that the fire comes at a level not higher than the customary grate level. In all cases, of course, the grates must be removed. An early practice of mounting nozzle-type burners through the firedoors of boilers should be avoided. Oil burns best in an environment of glowing firebrick; hence the firebox which encloses the burning oil logically belongs below the water-heating boiler surfaces. If

the refractory is placed against boiler heating surface, the boiler's capacity is reduced. Only in the case of Scotch Marine boilers, and certain types of sectional cast-iron boiler in which the water leg is dropped clear to the floor, is it permissible or usually necessary to burn the oil within the water-containing parts of the boiler. Ordinarily, hotter environment, with consequent cleaner combustion, will be obtained by setting the burner in the boiler ashpit.

TABLE XXXIV.—COMBUSTION-CHAMBER DIMENSIONS FOR ROUND BOILERS

Firing rate, g.p.h.	Dimensions, in.			
	A	B	C	D
1.3	13	12	11	6
1.7	14	14	14	6
2.3	15	16	16	6.5
2.7	16	18	16	6.5
3.0	17	18	18	7
3.5	17	20	18	7
4.0	18	20	20	7.5
4.5	18	22	20	7.5
5.0	19	22	22	7.5

Fuel oil must be burned in suspension. That is why more or less elaborate mechanisms have been created, and called oil burners. But the burner can at best prepare the oil for combustion and sometimes initiate combustion. The actual combustion process, however, occurs within the boiler firebox. So if the oil is really to burn in suspension, then the first requisite is that all solid objects, such as floor and walls, be out of the path of the oil spray. This means that the nozzle must be set high enough off the floor, and the walls of the firebox must be so placed, that the oil spray will enter the firebox, and the combustion process proceed well toward completion, without physical contact of the fuel and the environment.

FIREBOX DESIGN

A great deal has been said and written about the shape of the box. As a matter of fact, the shape is far less important than the size. So long as the walls are placed out of range of the oil so that impingement does not occur, the shape of the box can

follow generally the form of the boiler itself, with some modifications as shown below. But a good deal of thought should be given to the size of the box. It is customary to discuss size in

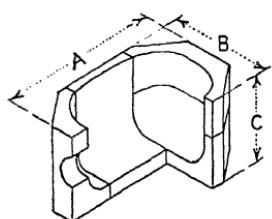


FIG. 163.—Ready-built firebox for gun-type burners. See Table XXXV for values of indicated dimensions.

terms of the relationship between combustion volume and gallons or pounds of oil burned per hour, or in terms of heat release, or again in terms of boiler output expressed in boiler horsepower or square feet of steam radiation or hot-water radiation. All these methods of expression are really the same and are readily convertible, one to the other. Perhaps the clearest relationship is the expression of B.t.u. released per cubic foot of combustion space per hour.

It has been said that all generalizations are dangerous. Equally trite is the statement "the exception proves the rule." As a matter of fact, it is useless to set down a code of rules to govern firebox design if the exceptions to the rules will be frequent or significant. Perhaps it will be safer to say that all available information and experience on the subject merely establish certain averages or limiting values, but we must clearly under-

TABLE XXXV.—DIMENSIONS FOR READY-BUILT FIREBOXES SHOWN IN FIG. 163

Size, gal./hr.	A, in.	B, in.	C, in.
1½	19	14½	12
2	21 $\frac{1}{16}$	14½	12
2½	21 $\frac{1}{2}$	18½	12
3	24 $\frac{3}{16}$	18½	12

stand that averages or average ranges of values are not necessarily indicative of what is rigidly correct. At best, these values only point out a tendency. Unfortunately, in the case of oil-burner combustion chambers, there has not been sufficient experimental and test work recorded to establish the validity of the accepted standards or averages. And perhaps, also, there never will be a scientifically correct set of values established, for it can readily be seen that what may be considered the desideratum depends entirely upon point of view. For instance, a certain design

formula or ratio may result in exceedingly high flame temperatures, which ordinarily would seem to be desirable, since all heat transfer is a function of temperature difference so that high flame temperature should mean higher rates of heat transfer into the boiler. But higher flame temperature also means shorter refractory life. So at least some check should be made on increased boiler efficiency against increased firebrick cost resulting from the higher flame temperature of a certain box design. In heating a small residence, 2000 gal. might be an average year's fuel consumption, and the average life of a firebox might be four years. Making the combustion volume less might increase the flame

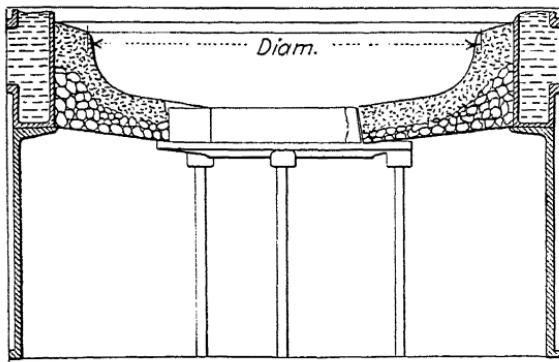


FIG. 164.—Hearth for vertical rotary-atomizing domestic burner. See Table XXXVI for data.

temperature by 200° and decrease the fuel consumption over a season by 3 per cent or 60 gal., but also might burn out the box in a single season. At 6 cts. per gal. the fuel saved is worth \$3.60, but the unused three years of life from the box might mean a loss of about \$10, so apparently the economy has been negative. The factors that must enter into consideration of firebox utility are numerous: cost of refractory, life of refractory, efficiency of boiler and combustion, cost of fuel, freedom from attention for cleaning boiler tubes, etc. It is impossible to resolve all these factors into a single formula which will give an answer by inserting values for the variables. So with this reservation in mind, the discussion of firebox volumes can be opened.

Because boiler efficiencies vary, heat release should be expressed on actual amount of fuel burned rather than on amount of useful heat absorbed by the boiler. For domestic and moderate-sized commercial boilers the rate of heat release should be about

30,000 B.t.u. per hr. per cu. ft. of combustion space; for larger heating boilers this can be increased up to 40,000. By combustion space is meant the total free volume in which combustion can occur and in which the gases of combustion can expand and undergo turbulence before entering flue passages or striking against water tubes. In small residential boilers or furnaces only about 20 per cent of this total volume will be enclosed within refractory, the remainder will be up within the water-jacketed walls of the boiler or the air-backed walls of the furnace. As the

TABLE XXXVI.—DIMENSIONS AND DATA FOR HEARTHS SHOWN IN FIG. 164*

Atomizer diameter, in.	Bowl diameter, in.	Boiler rating		Radiation load	
		Steam, sq. ft.	Hot water, sq. ft.	Steam, sq. ft.	Hot water, sq. ft.
3	16	750	1200	500	800
3	18	1125	1800	750	1200
3	22	1500	2400	1000	1600
3	24	1875	3000	1250	2000
4	26	2250	3600	1500	2400
4	28	2625	4200	1750	2800
4	30	3000	4800	2000	3200
4	32	3375	5400	2250	3600
4	34	3750	6000	2500	4000
4	35	4125	6600	2750	4400
4	36	4500	7200	3000	4800

* Courtesy Automatic Burner Corporation.

size of the unit increases, the percentage of combustion space enclosed within refractory will increase. Naturally, the higher the percentage of refractory-lined space, the better will be combustion, but this percentage cannot be satisfactorily increased by extending the firebrick walls up and thus shutting off boiler surface from exposure to the fire. The proper way to secure additional refractory surface is to pit the boiler setting and then to set the burner lower. In general, rectangular sectional or steel boilers permit of more refractory area than do round-vertical boilers.

The accompanying tables give dimensions for some typical fireboxes in conventional boilers and furnaces. These data apply to so-called conversion installations, *i.e.*, in which the burner is

to replace previous hand-fired or stoker-fired coal. Boilers which are originally designed for oil firing will usually have a combustion chamber provided by the manufacturer of the equipment. It will be found that the indicated fireboxes give a total combustion volume ranging from 30,000 to 40,000 B.t.u. release per cu. ft., or about 3.0 to 4.5 cu. ft. per gal. of oil burned per hour, or from 0.7 to 1.1 cu. feet per 100 sq. ft. of steam radiation output, or from 1.0 to 1.5 cu. ft. per boiler hp. developed. For the smaller, especially round-vertical, boilers, only 20 per cent of the total volume may be refractory-lined, but the percentage will increase

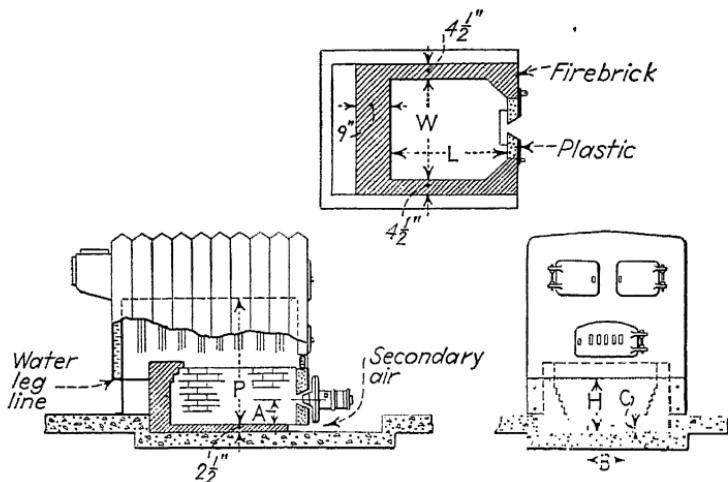


FIG. 165.—General construction of fireboxes for cast-iron sectional boilers. See Table XXXVII for data.

up to as much as 70 or 80 per cent in large commercial heating boilers.

If the firebox be made much smaller than here indicated, any of several ill effects will be experienced: it may be difficult to carry enough fire without smoking or forming carbon deposits, or the firebox temperature may become excessive with consequent fusing of the firebrick, or the backwall especially may wear away rapidly because of flame impingement. Should the firebox be made appreciably larger, on the other hand, the refractory will not heat up quickly enough, resulting in wasteful incomplete combustion, possibly accompanied by white fumes of unpleasant odor that will be expelled from the chimney.

TABLE XXXVII.—DATA FOR COMBUSTION CHAMBERS FOR CAST-IRON SECTIONAL BOILERS

<i>W</i> width of firebox, in.	Rating Steam, sq. ft.	G.p.h. Water, sq. ft.	<i>A</i> nozzle to floor, in.	<i>B</i> width secondary air port, in.	<i>C</i> height secondary air port, in.	<i>H</i> height boiler setting, in.	<i>L</i> length firebox, in.	<i>P</i> height combustion chamber, in.	Combust- ion volume, cu. ft.	% Volume within refractory
10	350	660	0.9	6.5	2.0	12	16	32	4.0	37.3
10	350	660	0.9	6.5	2.0	12	16	36	4.0	32.0
10	500	800	1.3	6.5	3.4	2.0	24	32	5.7	37.3
10	600	800	1.3	6.5	3.4	2.0	24	36	5.7	32.9
16	800	1280	2.0	6.5	4.5	2.0	14	22	9.2	40.0
18	800	1280	2.0	6.5	4.5	2.0	14	18	40	36.0
16	1000	1600	2.5	6.5	4.5	2.5	14	27	35	40.0
18	1000	1600	2.5	6.5	4.5	2.5	14	22	40	31.4
17	1250	2000	3.15	7.5	4.5	3.0	14	28	40	36.0
19	1250	2000	3.15	7.5	4.5	3.0	14	29	36	34.3
23	1500	2400	3.75	7.5	4.5	3.5	14	31	35	17.1
16	1500	2400	3.75	7.5	4.5	3.5	14	43	33	17.1
23	2000	3200	6.0	8.0	4.5	4.5	14	41	35	22.9
16	2000	3200	6.0	8.0	4.5	4.5	14	56	33	22.9
23	2500	4000	6.25	8.0	5.5	4.5	16	64	33	28.0
19	2500	4000	6.25	8.0	5.5	4.5	16	60	35	42.5
23	3000	4800	7.5	8.5	7.0	4.5	18	54	38	40.0
25	3000	4800	7.5	8.5	7.0	4.5	18	50	40	34.3
29	3000	4800	7.5	8.5	7.0	4.5	18	51	41	34.3
23	3500	6600	8.75	9.0	8.0	4.5	18	62	40	40.0
25	3500	6600	8.75	9.0	8.0	4.5	18	58	40	45.0
29	3500	6600	8.75	9.0	8.0	4.5	18	52	40	40.0
23	4000	6400	10.0	9.5	9.0	4.5	18	71	40	45.0
25	4000	6400	10.0	9.5	9.0	4.5	18	67	40	45.7
35	4000	6400	10.0	9.5	9.0	4.5	18	49	41	45.7
25	5000	8000	12.5	10.0	11.0	4.5	18	83	40	45.0
29	6000	8000	12.5	10.0	11.0	4.5	18	74	40	45.0
39	6000	8000	12.5	10.0	11.0	4.5	18	58	40	45.0
25	6000	9600	15.0	10.5	13.5	4.5	18	110	43	68.7
35	6000	9600	15.0	10.5	13.5	4.5	18	77	44	42.0
39	6000	9600	15.0	10.5	13.5	4.5	18	44	41.0	45.0
35	7000	11200	17.5	10.5	15.5	4.5	18	70	43	68.7
39	7000	11200	17.5	10.5	15.5	4.5	20	90	44	45.5
35	8500	13600	21.25	11.0	19.0	4.5	20	82	43	80.0
39	8500	13600	21.25	11.0	19.0	4.5	20	108	44	97.2
68	10000	16000	25.0	11.5	22.0	4.5	20	100	43	85.8

The shape of the box is, of course, a factor to be considered. The designs here indicated are intended to direct as much heat as possible, by reflection and radiation, from the refractory up to the boiler surfaces, as well as from the flame itself up into the boiler. That part of the boiler which absorbs heat from the hot gases is usually expected to absorb heat at a rate of about 3350 B.t.u. per hr. per sq. ft. of surface (having water on one side and hot gases on the other). But boiler surface directly exposed to flame or incandescent firebrick may absorb heat at a rate five to ten times as great.

In all cases of the firebox shapes shown, the walls are laid out to increase turbulence and avoid stagnant pockets in the corners. The heights of nozzles above floors are minimum that will assure freedom from impingement that might cause carbon formation on the floors.

In each case where secondary air port is shown, the percentage of air usually induced by natural draft is about 85 per cent. In most cases, the introduction of secondary air is made as direct and simple as possible. Elaborate checkerworks under floors, and long and tortuous channels for conducting secondary air into a multiplicity of ports scattered along the path of the flame, are to be avoided. It has been found that little or no advantage is thereby secured by enhanced intimacy of oil-air mixture, or preheating of air by hot brick, etc. On the contrary, these indirect air ducts merely create additional friction in the path of air flow, requiring more draft to bring the air into the firebox. Sooner or later these checkers and ducts cave in or fill up with muck, so that the air flow is retarded.

The total free area of secondary air port is governed by the percentage of total air taken in as secondary, and the intensity of available draft. For burners which take in about 15 per cent of total air as atomizing or primary air, the remaining 85 per cent is induced by natural draft. For ordinary heating-plant draft conditions (ranging around 0.1 in. of water), there will be required about $1\frac{1}{2}$ sq. in. of free area per boiler horsepower or about 1 sq. in. per 100 sq. ft. of equivalent steam radiation boiler capacity. Of course, where a greater percentage of primary air is used and is delivered into the firebox by fan, the required area can be diminished accordingly. Secondary air ports should always be made larger, whenever there is a question as to size, since it is always

possible to reduce the opening when the combustion test reveals excess air. It is sometimes difficult, on the other hand, to increase an air-port opening once the firebox is built, should it prove too small.

Since an important function of the firebox is to protect those parts of the boiler which might be injured by heat, these designs take into account the covering of mud legs and rivet seams which are close to the flame, as well as grate lugs and other parts which are not water cooled. Where the burner installation calls for

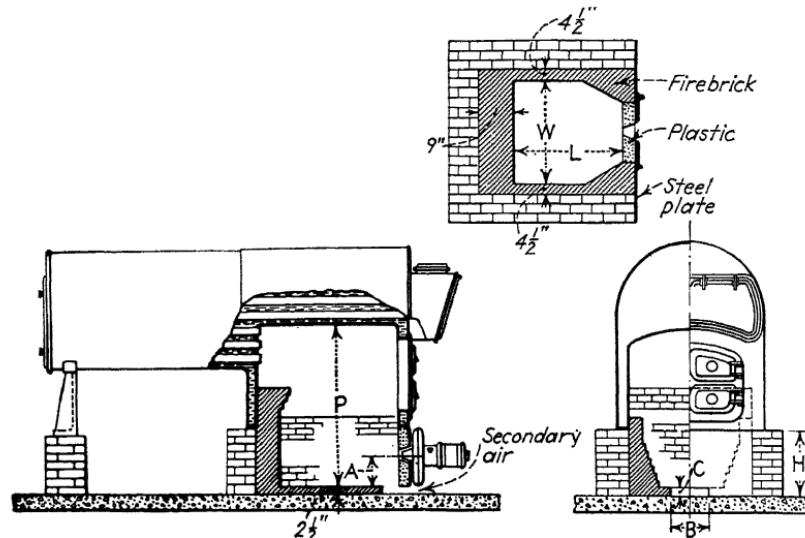


FIG. 166.—General construction of fireboxes for steel-firebox boilers. See Table XXXVIII for data.

metallic supports on the boiler front or under the hearth, these parts also are given refractory protection.

In many cases the use of an insulating layer of silicious or other type of nonconductor is advisable, although not mandatory. Usually, these special insulators will earn their investment many times over during the life of the installation by preventing the flow of heat to and through the boiler setting. Where it is intended or required that such special insulators be used, the additional cost should be included in the sales price.

As a rule, back firing of conventional low-pressure boilers does not enhance operating efficiency, but on the contrary may contribute to poor operating results by virtue of flame short circuit-

TABLE XXXVIII.—DATA FOR COMBUSTION CHAMBERS FOR STEEL-FIREBOX BOILERS

<i>W</i> width of firebox, in.	Rating Steam, sq. ft.	G.p.h. Water, sq. ft.	<i>A</i> nozzle to floor, in.	<i>B</i> width secondary air port, in.	<i>C</i> height secondary air port, in.	<i>H</i> height boiler setting, in.	<i>L</i> length firebox, in.	<i>P</i> height combustion chamber, in.	Combust- ion volume, cu. ft.	% Volume within refractory
15	1000	1600	2.5	6.5	4.5	2.5	14	49	11.4	28.6
21	1500	2400	3.8	6.5	4.5	3.5	14	22	17.1	26.9
21	2000	3200	5.0	7.0	4.5	4.5	14	30	52	22.9
21	2800	4000	6.3	7.0	5.5	4.5	14	39	50	28.6
21	2800	4000	6.3	7.0	5.5	4.5	14	39	50	28.0
27	2800	4000	6.3	8.0	5.5	4.5	14	28	55	28.6
27	3300	5280	8.0	8.5	7.5	4.5	18	35	59	37.7
27	4000	6400	10.0	8.5	9.0	4.5	18	41	69	45.7
33	4000	6400	10.0	8.5	9.0	4.5	18	37	62	46.7
33	6000	8000	12.3	9.0	11.0	4.5	18	41	62	57.1
33	9600	9600	15.0	9.5	13.5	4.5	18	51	62	68.7
39	6000	9600	15.0	9.5	13.5	4.5	18	41	67	68.7
39	7500	12000	18.8	10.5	16.5	4.5	20	53	64	85.9
39	7500	12000	18.8	10.5	16.5	4.5	20	50	69	85.9
39	8500	13800	18.8	10.5	16.5	4.5	20	56	69	85.9
44	8500	13600	21.3	10.5	19.0	4.5	20	47	74	97.2
39	10000	16000	25.0	11.0	22.0	4.5	20	49	69	86.8
44	10000	16000	25.0	11.0	22.0	4.5	20	41	74	85.8
44	12500	20000	31.3	11.5	28.0	4.5	20	55	69	107.2
60	12500	20000	31.3	11.5	28.0	4.5	20	58	79	107.2
60	16000	24000	37.5	12.0	33.0	4.5	22	55	74	128.8
50	15000	24000	37.5	12.0	33.0	4.5	22	51	81	128.8
50	18500	26600	46.3	12.5	31.0	6.0	25	65	77	158.6
56	18500	26600	46.3	12.5	31.0	6.0	25	67	79	158.6
66	18600	26600	46.3	12.5	31.0	6.0	25	61	89	158.6
54	21000	33600	52.3	13.0	35.0	6.0	25	72	80	180.0
62	21000	33600	52.3	13.0	35.0	6.0	25	58	80	190.0
66	24000	38400	60.0	14.0	40.0	6.0	30	75	85	205.8
62	24000	38400	60.0	14.0	40.0	6.0	30	67	85	205.8
62	30000	48000	75.0	16.0	50.0	6.0	30	84	85	257.0
68	30000	48000	75.0	16.0	50.0	6.0	30	74	88	257.0

ing. When a burner flame must buck the draft by attempting to move in a direction opposite to the normal flow created by the draft through the boiler, it cannot be expected that either the fire or the boiler will work effectively. It is far better to allow all operations to occur in normal direction and sequence than to try to secure additional flame travel by disturbing these factors.

REFRACTORIES

The most commonly used refractory for ordinary oil-burner installations is standard firebrick, which is made from the better grades of fire clay (aluminum silicate), and which comes in the standard size of 9 by $4\frac{1}{2}$ by $2\frac{1}{2}$ in. as well as many special shapes. The standard firebrick weighs about 8 lb., contains about 100 cu. in. and runs 17 brick to the cubic foot, weighing about 136 lb. per cu. ft. Laid $4\frac{1}{2}$ in. thick, 1 sq. ft. of wall contains 7 brick. This brick is satisfactory for temperatures not exceeding 2500 to 2700°F. The best bond for setting ordinary firebrick is one of the air-setting high-temperature cements, which should be mixed with water to a consistancy of a thick cream, into which the brick should be dipped. After dipping, the brick should be laid into the wall, each brick being firmly pressed or tamped into place. Thick clay laid on with a trowel, for binder, is not recommended. After the box is built, the entire surface should be painted with the mixed cement so as to fill up all cracks and crevices. Since it is at the exposed edges that wear first occurs, the brick should be carefully handled to prevent chipping, and the wall surface should be smooth.

While standard firebrick serves very well in most heating-plant installations, it has the disadvantage of possessing great weight. Therefore every time the burner stops and the box cools off, a considerable amount of heat is dissipated, which must again be furnished and absorbed by the brick when the burner next starts. Aside from wasting a fair amount of heat, this has the disadvantage of requiring that the burner operate for some time in a cold firebox. To overcome this undesirable feature of operation in intermittent control of oil burners, special refractories have been developed. These have very low specific gravities: *i.e.*, they are much lighter than fire-clay brick. The heat conductivity is also lower, so that a thinner wall will have the same insulating qualities. A firebox constructed of such material will

come up to temperature much more rapidly, and will require less heat to do so. These special refractories come in a variety of forms and are made of several different substances. Some can be had as ready-made bricks of either standard dimensions or of half thickness. Others come as powder which must be mixed with water and then poured, much as concrete is, into a form which has been constructed inside the boiler. When it hardens, this forms a one-piece firebox. It is also possible to purchase

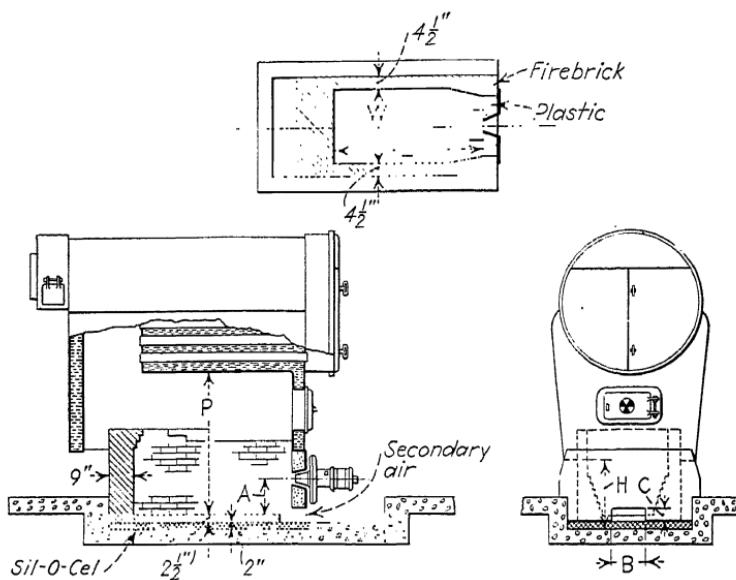


FIG. 167.—General construction of fireboxes for welded-steel tubular boilers.
See Table XXXIX for data.

precast chambers, which can be fitted into conventional boiler fireboxes.

In the appendix will be found technical and practical data pertaining to refractories as usually employed in combustion practice. The following notes are explanatory of the data which has been presented in this chapter on fireboxes for cast-iron sectional boilers, steel-firebox boilers, and welded-steel tubular boilers in Tables XXXVII, XXXVIII and XXXIX.

Combustion volume was computed as an approximation of the free space enclosed between firebox floor, crown sheet, boiler front and bridge wall. The width is taken as W plus $4\frac{1}{2}$ in.

TABLE XXXIX.—DATA FOR COMBUSTION CHAMBERS FOR WELDED-STEEL TUBULAR BOILERS

<i>W</i> width of firebox, in.	Rating Steam, sq. ft.	G.p.h. Water, sq. ft.	<i>A</i> nozzle to floor, in.	<i>B</i> width secondary air port, in.	<i>C</i> height secondary air port, in.	<i>H</i> height boiler setting, in.	<i>L</i> length firebox, in.	<i>P</i> height combustion chamber, in.	Combust- ion volume, cu. ft.	% Volume within refractory
20	1500	2400	3.75	4.5	3.5	14	34	35	17.1	40.0
20	1750	2800	4.75	6.5	4.5	14	40	35	20.0	40.0
16	2000	3200	6.0	7.0	4.5	14	55	35	22.9	40.0
21	2300	3680	6.75	7.0	6.0	14	51	35	26.3	40.0
21	2600	4160	6.5	7.0	6.0	14	57	35	29.7	40.0
21	2800	4480	7.0	7.5	6.5	18	62	39	32.0	46.1
21	3100	4960	7.75	8.0	7.0	14	62	39	36.4	46.1
21	3400	5440	8.5	8.0	7.5	18	68	39	38.9	46.1
21	3600	5760	9.0	8.5	8.0	18	75	37	41.2	48.6
26	3600	5760	9.0	8.5	8.0	18	57	41	41.2	43.9
27	4000	6400	10.0	8.5	9.0	18	61	41	45.7	43.9
27	4500	7200	11.25	8.5	10.0	18	69	41	51.4	43.9
27	5300	8320	13.0	9.0	12.0	18	83	39	59.5	46.1
27	5300	8320	13.0	9.0	12.0	18	80	41	59.5	43.9
27	6000	9600	15.0	10.5	13.5	18	91	41	68.7	43.9
33	6000	9600	15.0	10.5	13.5	18	75	42	68.7	42.8
33	6500	10400	16.25	10.5	14.5	18	82	43	74.3	41.8
33	6500	10400	16.25	10.5	14.5	18	81	42	74.3	42.8
33	7000	11200	17.5	11.0	16.5	18	84	44	80.0	45.5
33	7000	11200	17.5	11.0	16.5	20	84	44	80.0	45.5
39	9500	15200	23.75	11.0	21.0	4.5	20	73	44	81.5
39	9500	15200	23.75	11.0	21.0	4.5	20	65	50	81.5
44	12000	19200	30.0	11.5	26.0	4.5	20	73	50	102.7
44	12000	19200	30.0	11.5	26.0	4.5	22	64	57	102.7
49	14500	23200	36.25	12.0	32.0	4.5	22	80	50	124.5
52	14500	23200	36.25	12.0	32.0	4.5	22	70	54	124.5
60	17000	27200	42.5	12.5	38.0	4.5	22	89	52	42.3
52	17000	27200	42.5	12.5	38.0	4.5	22	83	54	145.9
55	20000	32000	50.0	13.0	33.0	6.0	25	90	55	40.7
60	20000	32000	50.0	13.0	33.0	6.0	25	79	68	171.4
60	25000	40000	62.5	14.0	42.0	6.0	30	90	64	43.1
61	25000	40000	62.5	14.0	42.0	6.0	30	86	66	214.5
69	30000	48000	75	16.0	50.0	6.0	30	92	66	45.5
69	30000	48000	75	16.0	50.0	6.0	30	66	287.3	45.5

The height is taken as P , and the length as L . The product of these three therefore gives combustion volume.

On boilers rated up to 8500 sq. ft. of steam radiation for steel-firebox boilers and cast-iron sectional boilers, and up to 7000 sq. ft. of steam radiation on welded-steel tubular boilers, the heat release is figured at 30,000 B.t.u. per cu. ft. of combustion volume. Above these values, the heat release is figured at 40,000 B.t.u. per cu. ft. An over-all efficiency of 70 per cent was assumed, so that the required combustion volume is derived from the expression:

$$V = \frac{\text{sq. ft. steam radiation} \times 240}{0.7 \times 30,000 \text{ (or } 40,000\text{)}}$$

Secondary air-port areas are computed on basis of 1 sq. in. of opening per 100 sq. ft. of steam radiation.

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CHAPTER XV

INDUSTRIAL BOILER-PLANT INSTALLATIONS

There are three types of oil burners applicable to moderate and large steam-generating plants: mechanical pressure-atomizing, horizontal-rotary and air or steam atomizing. The installation methods for all three are in general similar, as regards furnace construction, oil preheating, pumps and piping and controls.

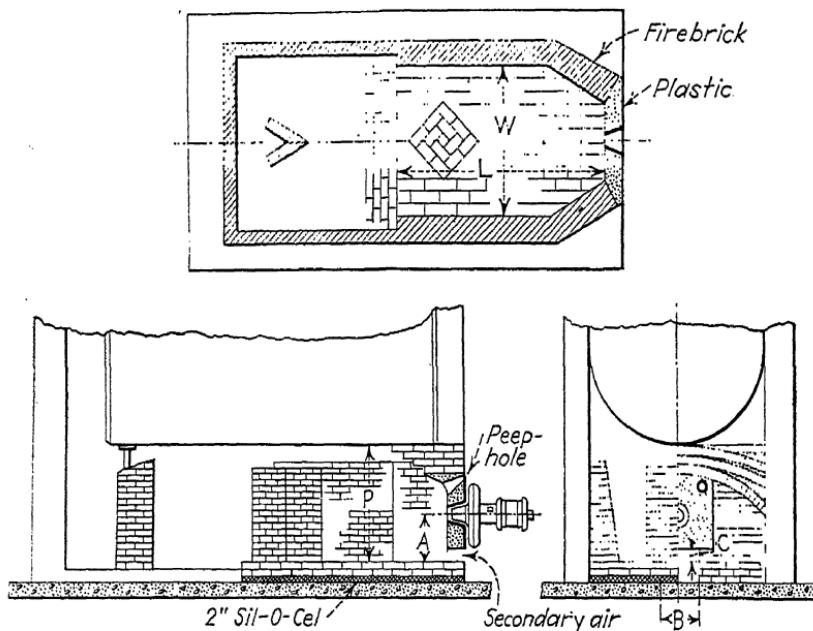


FIG. 168.—General construction of furnaces for horizontal return-tubular boilers.
See Table XL for data.

What differences might be necessary in the installation of the various types have already been discussed in Chaps. VIII, IX, XII and XIII. In Chap. XIV was presented a discussion of firebox construction for heating boilers. This chapter will review the principles underlying combustion-chamber design for large boilers. It will also show usual practice in respect to control apparatus. Finally, there will be given a detailed investigation of combustion control systems and their applications.

TABLE XL.—DATA FOR COMBUSTION CHAMBERS FOR HORIZONTAL RETURN-TUBULAR BOILERS

<i>W</i> width of firebox and diameter of boiler, in.	Boiler hp.	G.p.h.	<i>A</i> nozzle to floor, in.	<i>B</i> width second- ary air port, in.	<i>C</i> height second- ary air port, in.	<i>P</i> height com- bus-tion cham- ber, in.	Length of boiler, ft.	<i>L</i> length of fire- box, in.	Com- bus-tion vol- ume, cu. ft.
60	80	26.6	14.0	25	4.5	50	16	67	127.5
60	90	30.0	15.0	28	4.5	50	18	83	143.5
66	100	33.3	16.0	24	6.0	54	16	77	159.5
66	110	36.6	16.5	26	6.0	54	18	85	175.5
72	125	41.6	17.5	30	6.0	58	16	83	200.0
72	150	50.0	18.0	35	6.0	60	18	96	239.5
72	165	55.0	19.0	39	6.0	66	20	96	263.5
78	180	60.0	20.0	42	6.0	72	18	98	287.0
78	200	66.6	21.0	47	6.0	78	20	90	319.0

NOTE: Combustion volumes in this table are based upon boiler operation at 100 per cent of rating, with heat release of 30,000 B.t.u. per cu. ft. of volume.

At higher ratings, heat release will be proportionately higher: at 125 per cent rating heat release rate will be 37,500 B.t.u. per cu. ft.; at 150 per cent rating, it will be 45,000 B.t.u. per cu. ft. of combustion volume.

By combustion volume is meant $W \times L \times P$. Heat release is computed as

$$\frac{\text{boiler hp.} \times 33,500}{0.7 \times 30,000}$$

Secondary air-port area is calculated on basis of 1.4 sq. in. of area per boiler hp.

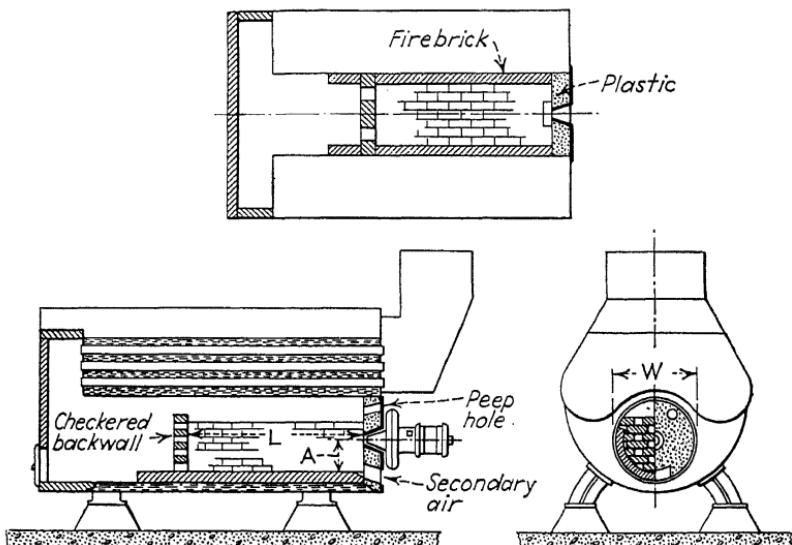


FIG. 169.—General construction of furnace in Scotch Marine boiler. See Table XLI for data.

TABLE XLI.—DATA FOR COMBUSTION CHAMBERS FOR SCOTCH MARINE BOILERS

<i>W</i> diameter of firebox, in.	Rating		Diam- eter of shell, in.	Length of tubes, in.	A nozzle to floor, in.	<i>B</i> × <i>C</i> area of second- ary air port, sq. in.	<i>L</i> length of firebox, in.	Com- bus- tion volume, cu. ft.
	boiler hp.	G.p.h.						
26	20	6.6	44	78	10	28	62	19.1
26	25	8.3	44	96	11	35	78	23.9
26	30	10.0	48	96	11	42	90	28.6
26	35	11.6	48	114	12	49	108	33.4
30	40	13.3	54	96	12	56	90	38.2
32	50	16.6	60	114	13	70	102	47.8
36	60	20.0	72	114	14	84	98	57.2
36	75	25.0	72	144	15	105	122	71.5
42	100	33.3	84	160	16	140	120	95.5
45	125	41.6	90	160	17.5	175	130	114.2
45	150	50.0	90	174	18	210	155	143.2
48	175	58.3	96	174	20	245	160	167.0
52	200	66.6	102	174	21	280	155	191.0

NOTE: Combustion volumes in this table are based upon boiler operation at 100 per cent rating, with heat release of 50,000 B.t.u. per cu. ft. of volume. At 125 per cent rating, heat release would be at rate of 62,500 B.t.u. per cu. ft. If boiler is to be operated at rating much above 125 per cent, it is advisable to extend the firebox length by calculating *L* at a maximum heat release of 65,000 B.t.u. per cu. ft. of combustion volume. The formula for calculating *L* when the heat release is 65,000 B.t.u. is

$$\frac{\text{boiler hp.} \times 33,500}{0.55 \times \frac{W^2}{W^2} \times 65,000} \quad \text{or} \quad \frac{\text{boiler hp.} \times 0.94}{W^2}$$

Secondary air-port area is calculated on basis of 1.4 sq. in. per boiler hp. If the total area cannot conveniently be placed below the burner, an auxiliary air port may be placed above the burner.

The accompanying drawings on horizontal return-tubular and Scotch Marine boilers show standard practice on combustion chambers, together with tabular data indicating the bases of design for various ratings. These boilers are generally operated on natural draft. Except for the mechanical pressure-atomizing burner, most burners operate best on these two boilers under either natural or induced draft; the mechanical pressure atomizer can be adapted to forced draft.

The mechanical pressure atomizer finds its principal application on water-tube boilers, and for high ratings, forced draft is frequently resorted to. In many such installations, air for combustion is forced by the fan through passages in the walls and floors, for the double purpose of cooling the refractories and pre-heating the air. Construction details are indicated in the

accompanying illustrations on water-tube installations of oil burners.

Most boilers operating under natural draft will have adequate draft for oil-burner operation when the stack induces an overfire static pressure of about 0.1 to 0.2 in. water below atmosphere. Under these conditions the secondary air port should be propor-

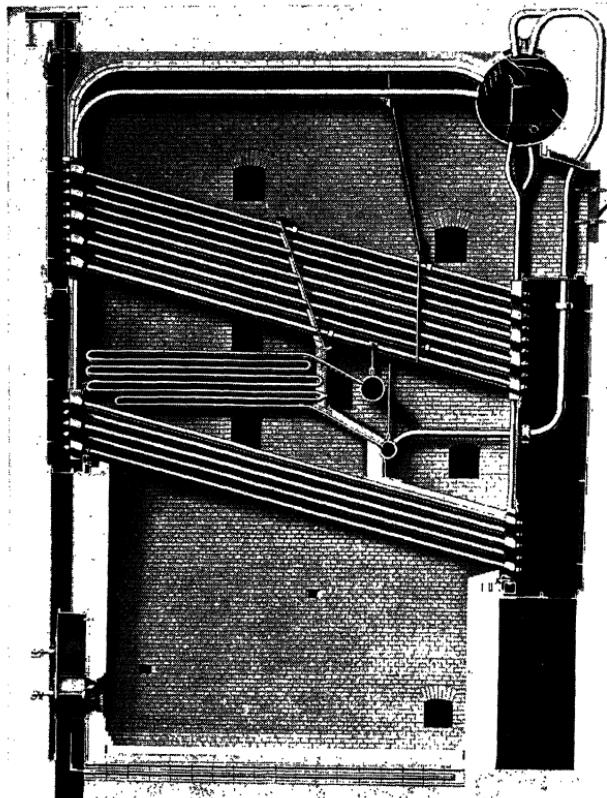


FIG. 170.—Water-tube boiler, oil-fired, under forced draft. (*Courtesy of Babcock and Wilcox Company.*)

tioned at 1.4 sq. in. per developed boiler hp. When air registers are used, as with mechanical pressure-atomizing burners, the secondary port cannot usually be designed so generously, the usual ratio then being about 0.8 sq. in. per developed hp. at maximum rating. Hence, with an air register, a somewhat greater draft intensity will be needed, ranging from 0.2 to 0.3 in. It is, of course, quite important, regardless of the available draft

(provided of course that it be greater than the minimum), that some means be available for altering the draft as required by combustion, the usual means being a damper in the stack or a damper in the secondary port or a shutter on the register if used.

When forced draft is employed, it will be found that air must be discharged from the fan into the supply duct at about 3 in. static pressure. After passing through the cooling passages in the

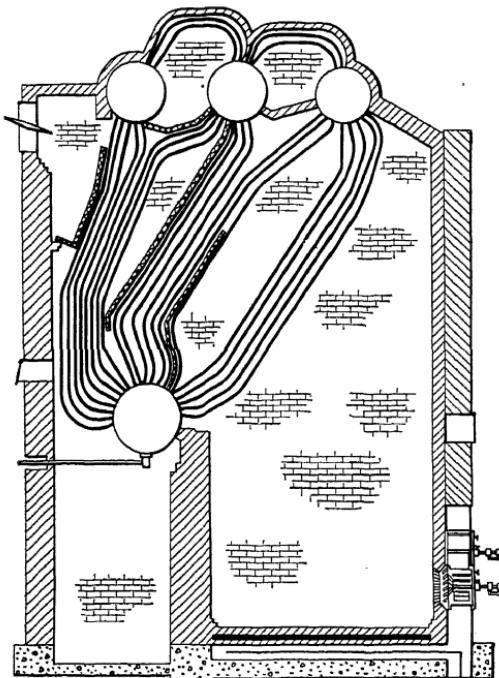


FIG. 171.—Stirling boiler, oil-fired, under forced draft with mechanical pressure-atomizing burners.

walls and floor, and when it is at the entrance to the register, the air pressure has usually diminished to less than 1 in. Whether the firebox pressure will be above or below atmosphere will be determined by the draft effect induced by the stack as balanced against the static resulting from the fan. Very seldom is actual pressure above atmosphere required in the firebox, except in marine work.

Combustion volume for water-tube boiler settings can in general follow the practice for fire-tube boilers, *viz.*, 30,000 B.t.u. per cu. ft. release per hr., at normal ratings. It is best to design for

that load condition which prevails during the greatest length of time of burner operation. In other words, should the load condition in any particular installation average around 150 per cent of boiler rating, then the combustion volume should be designed at about $1\frac{1}{2}$ cu. ft. per *rated* boiler hp. From this it will follow that at load conditions other than 150 per cent of rating, the

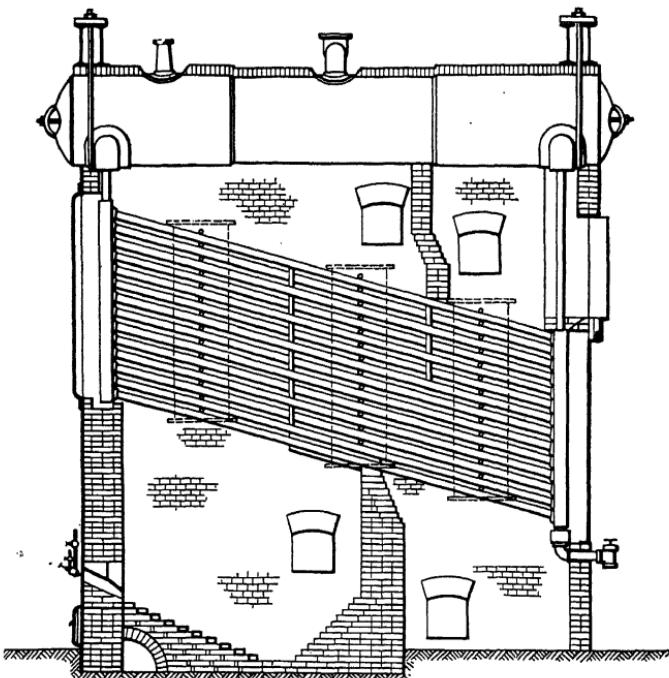


FIG. 172.—Water-tube boiler, under natural draft with air- or steam-atomizing nozzles.

heat release will be either greater or less than 30,000 according as the load is above or below 150 per cent rating.

Aside from correct sizing of the combustion volume the most important feature of a water-tube installation is the matter of baffling. In fact sometimes the baffling offers more difficulty in design than the problem of providing adequate combustion volume.

It can be accepted as a general principle that vertical baffling is to be preferred to horizontal baffling if the conditions of installation permit it. From this it will frequently follow that the oil burner itself should be placed at the rear of a boiler rather than

the front—by the front being meant that end at which the tubes are higher than the opposite end. The reason for this is that when firing from the front, with vertical baffles, the draft tends to pull the fire upward before it has reached the end of the furnace, whereas when firing from the rear the draft assists in pulling the fire the full length of the furnace.

If the construction of the boiler or its setting precludes rear firing, then it is sometimes preferable to use horizontal baffling,

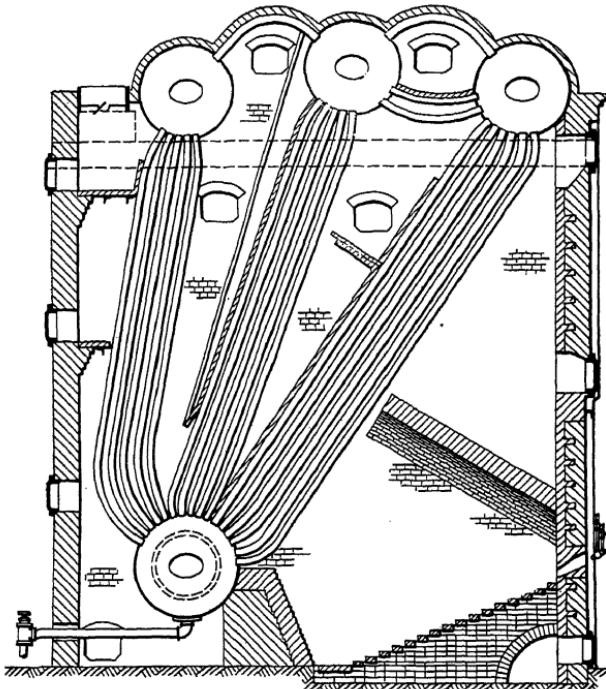


FIG. 173.—Stirling boiler, under natural draft with air- or steam-atomizing nozzles. (*Courtesy of The Denver Fire Clay Company.*)

although, of course, vertical baffling is sometimes employed. The use of a Dutch oven extending outward from the boiler front enhances operation with vertical baffling, as a bridge wall may then be constructed under the first baffle, and direct the flames upward into the first pass. The oven assures sufficient combustion volume, so that combustion may be complete before the flames strike the tubes.

If horizontal baffling must be used, the lower baffle should be placed above at least one row of tubes, and possibly three or four,

depending of course on the nature and construction of the boiler. When the baffle is placed below the first row of tubes, there is a tendency to create a bottled condition within the firebox, with a result that firebox temperatures become excessively high and slagging or fusing of the refractory results.

Regardless of the style of baffling or location of the burner, it is essential that combustion be practically complete before the flames enter the first pass. When this does not occur, and flames

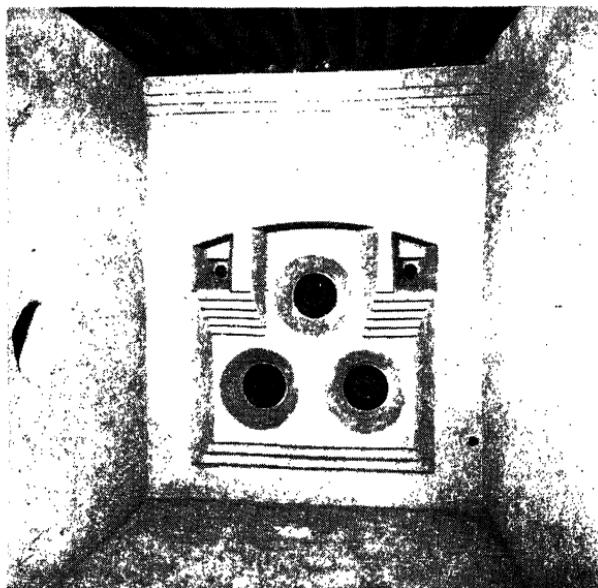


FIG. 174.—View of furnace front under water-tube boiler, with three burners.
(Courtesy of Mid-West Heat Service Company.)

impinge directly upon water tubes, failure of the tubes usually occurs. Aside from the obvious disadvantages of tube failure, such as shutdowns and cost of replacement, impingement also causes lowered combustion efficiency because of the chilling effect of the tubes on the fire.

Oil pumps and pumping, oil piping, preheating and other phases of oil handling have all been discussed in detail elsewhere in this book. There are no problems peculiar to power plants that will not be encountered in other aspects of oil-burner applications. The accompanying illustrations show good practice regarding the layout of the oil-handling systems for steam-generating plants.

The same remarks are true for the wiring of electric motors as used on pumps and burner motors. The subject of combustion control, however, is a fairly complex one, and must be considered in considerable detail. The remainder of this chapter is devoted to discussion of control systems and their applications.

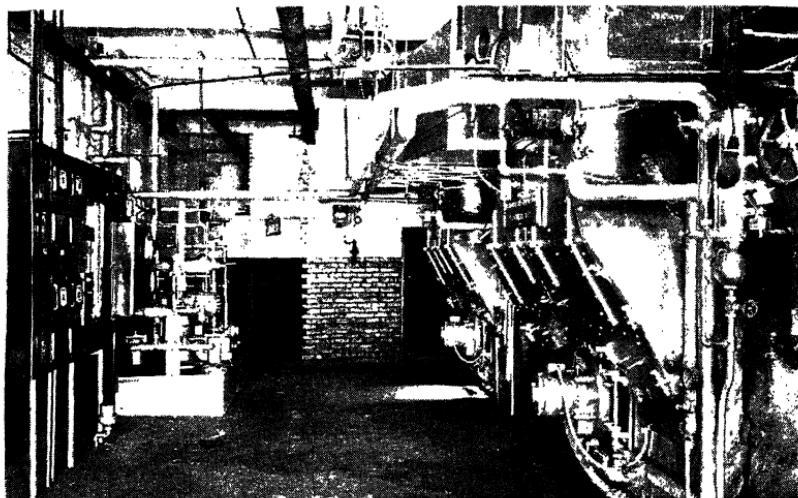


FIG. 175.—Scotch Marine boilers, fired with Todd horizontal-rotary burners, employing electric control system to modulate fire size.

COMBUSTION CONTROL SYSTEMS

There are many factors in power-plant installations which require regulation, and many methods have been devised for accomplishing control over one or more of these variables. Hence, it is found that in actual practice installations will be equipped in a manner ranging from strictly manual—with no instruments or control apparatus beyond simple hand-operated valves—to extremely elaborate systems designed to automatically compensate for all possible variable factors and to indicate and make records of operating performance.

Essentially, the primary object of any combustion control system is to maintain constant steam pressure, or to limit pressure variation as closely as possible, with minimum fuel consumption. An elementary control system would be one that merely proportioned oil feed to boiler pressure. This could be done inexpensively in at least three ways, a direct-acting oil valve actuated by steam pressure through a bellows or diaphragm, or a

reversing motor controller actuated by a pressurestat and in turn transmitting motion to an oil-valve stem or to an oil-pressure regulator, or two metering valves in conjunction with a solenoid valve whereby the flow through the metering valve set for high fire could be opened and shut through a pressurestat while the metering valve set for low fire would flow continuously.

Such an elementary system is simple and will maintain desired steam pressure. But almost without exception the resultant combustion would be inefficient, for this method gives no control whatever over air supply. It follows that a control system must synchronize air admittance in amount required by the oil. Practically, this is far from simple, for numerous reasons, some of them inherent to the combustion of any fuel, others peculiar to the burning of oil.

The problems to be solved for correct, efficient, and effective control of oil combustion are these:

1. Selection of a suitable device for proportioning oil flow to boiler load in a manner which will maintain required steam pressure, within permissible limits.
2. Controlling of stack draft to compensate for variations due to atmospheric conditions, such as wind direction and velocity and barometric pressure.
3. Synchronizing of both primary and secondary air to oil flow.
4. Maintenance of uniform oil pressure and compensation for variation in oil viscosity.
5. Compensation for changes in pressure drop through the boiler with changes in load conditions.

Even when all five of these factors are under adequate control, it is also necessary that air infiltration, because of leaks in the boiler setting, be prevented.

THE FIVE MAJOR FACTORS IN COMBUSTION CONTROL

Each of these five major points in combustion control can now be considered in detail.

1. Combustion Controllers.—There are available a large number of very satisfactory combustion controllers that are adaptable to fuel oil. These may be classified as electrically operated; fluid operated, using either air, oil, or water, under pressure; combination electric and fluid operated.

The simplest form of electric modulation consists of a pressure-stat having a small resistance coil similar to a Wheatstone bridge, connected to a reversing motor in such manner that variations in pressure are translated into electric impulses which will move the motor in the direction required to offset the pressure change.

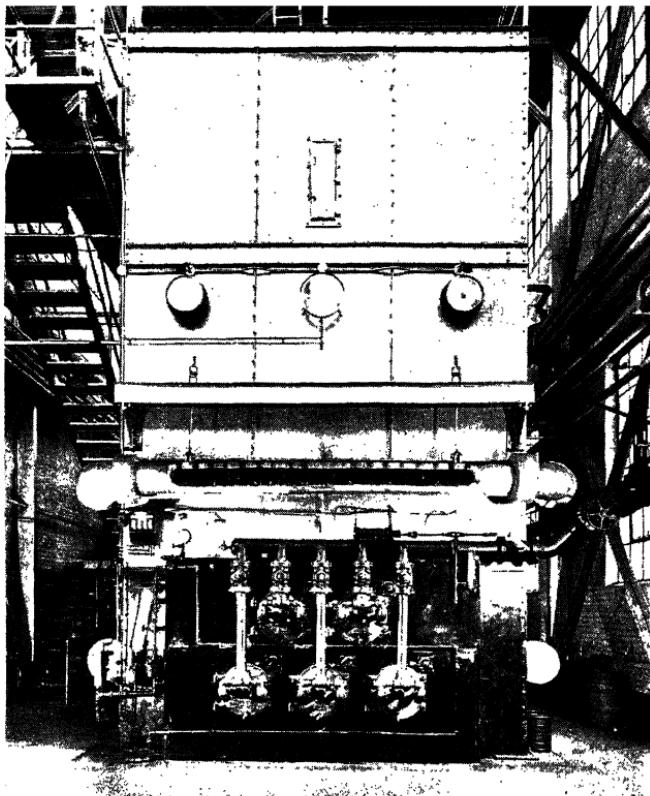


FIG. 176.—Todd combination oil-and-gas burners with hydraulic combustion control system. (*Courtesy of Mid-West Heat Service Company.*)

Rotation of the motor is communicated to an oil valve, and to air-admission ports on the boiler front or to an uptake damper, or to both. Figure 175 illustrates an installation of this type.

Fluid controllers are made in a variety of forms, using compressed air or oil under pressure or water under pressure. Boiler steam pressure is caused to operate a pilot valve, through a diaphragm or bellows or similar device, so that fluid is admitted into a cylinder fitted with a piston. The admission is directed

to that side of the piston which will cause it to move in the required direction for offsetting the pressure change which caused the pilot-valve motion. The piston is linked to the fuel-oil control valves, to the air inlets at the burner and to the uptake damper, the latter in combination with or in lieu of the former. By running a line shaft it is possible to have a single controller regulate a battery of boilers. This is not advisable if peak efficiency is sought, for each boiler may require its own individual draft setting to allow for slight differences in pressure drops through boilers, proximity to the stack, etc.

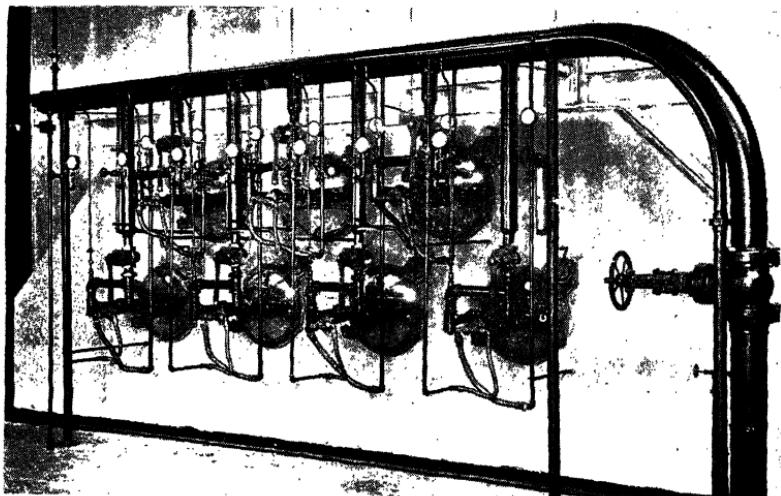


FIG. 177.—Todd combination oil-and-gas burners under water-tube boiler.

If sufficient water pressure (over 30 lb.) is available at all times and if the water is reasonably clear, this medium will do and it is the least expensive. If air pressure is available, it is more feasible and gives more accurate control. In larger plants, however, oil seems to yield the best all-round results; the usual method of application is to circulate it, under pressure, between a small reservoir and the control system. A light petroleum oil is used. Figure 176 shows such an installation.

2. Draft Controllers.—Commercial forms of draft-control apparatus may be classified as the balanced barometric type; and the overfire control type, subclassified as the pneumatic, hydraulic and electric types.

The balanced barometric draft controller is installed either on the breeching or at the base of the stack. If on the breeching, it must be located between the stack and the first boiler on the line. It consists essentially of a damper plate mounted on knife edges in a position parallel to the breeching or direction of flow of the gases, and it swings into a boxlike frame which is attached to the breeching over an opening in it which corresponds to the

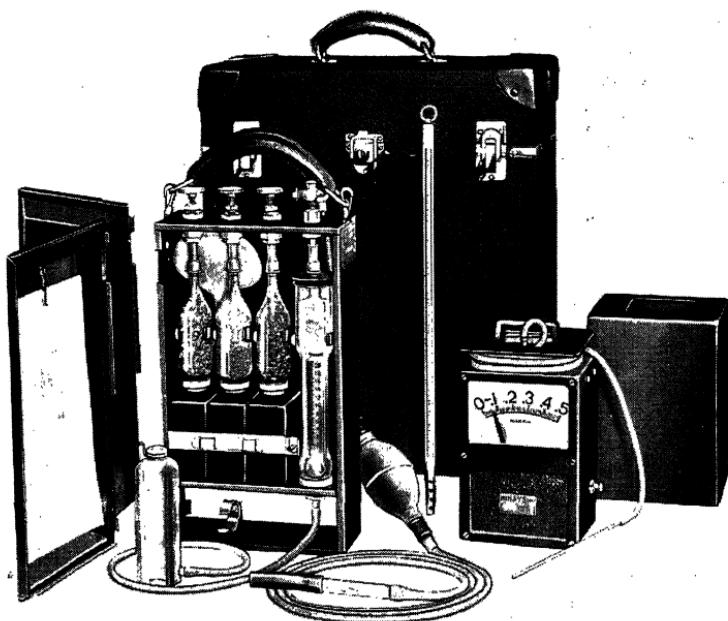
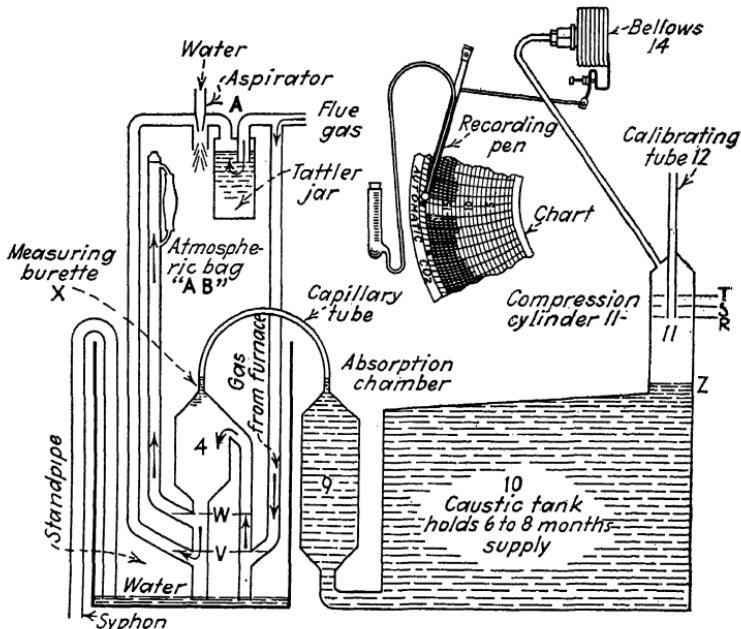


FIG. 178.—Portable combustion test apparatus, comprising Orsat, draft gauge, and stack thermometer. (*Courtesy of The Hays Corporation.*)

size of the frame. The damper plate is counterbalanced by an adjustable weight, whereby it can be caused to swing inwards and thus allow air to be drawn directly into the breeching from the boiler room. Thus at any predetermined intensity of draft in the stack or breeching the damper will check the draft by swinging inwards and so prevent the equivalent amount of air from being drawn through the firebox. While the balanced barometric controller is relatively inexpensive and offers low-cost operation from a maintenance viewpoint, it has two objections. First, it removes large quantities of warm air from the boiler

room, and so increases the radiation losses from the boiler and steam pipes. Second, it cannot and does not compensate for changes in draft requirements arising from sooted boiler tubes, from air leaks occurring in boiler settings, and from varying draft requirements resulting from varying load conditions, as regards a single boiler installation, and in the case of a battery it makes no provision for individual boiler-draft needs as regards pressure drops through separate boilers.



G. 179.—Diagram of continuous recording carbon dioxide analyser. (Courtesy of The Hays Corporation.)

In the overfire draft controller, a tube is inserted into the furnace through the boiler setting so as to communicate furnace pressure to a diaphragm which in turn can actuate either a piston (on pneumatic or hydraulic types) or a resistance coil of a motor. Whatever be the means, the resultant motion is employed continuously to vary the position of the uptake damper so that a uniform draft may be maintained within the boiler firebox.

3. Synchronizing Air to Oil Flow.—Care must be taken in linking the oil valve and damper control to compensate for the fact that the area of the path of air flow, rather than the linear

movement of the damper rod, must be proportioned to the effective opening in the oil valve. The curve showing the relation between inches of damper-rod movement against area of opening resulting is usually not a straight-line function, but rather a curved-line function. For successful control it is necessary to have straight-line operation. For accomplishing this there are various compensating cam arrangements available. However, it is usually necessary to design a cam for each separate installation.

4. Regulation of Oil Pressure and Compensation for Oil Viscosity.—It might be said that rigid control over oil pressure and

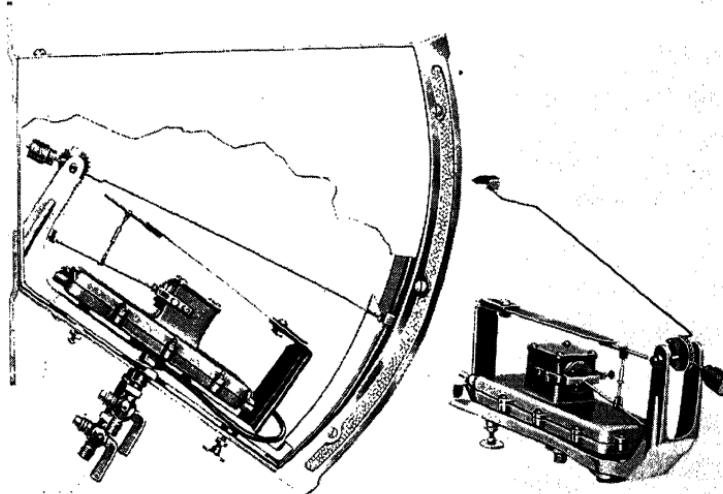


FIG. 180.—Differential draft indicator, to indicate draft loss between any two points in boiler setting. (*Courtesy of The Hays Corporation.*)

compensation for changes in oil viscosity offer the most troublesome features in oil combustion systems, or, in general, that regulation of oil flow presents knotty technical problems. Conventional valves will maintain uniform flow only if two factors, oil pressure and oil viscosity, are held within close limits. Obviously, for any position of the oil valve corresponding to a set of conditions in the controller, the amount of oil passed through a momentarily fixed orifice in the valve is determined by pressure and viscosity; with a fixed orifice the amount of oil will vary directly with the pressure and inversely with the viscosity. At the same time, of course, changes in pressure or viscosity would

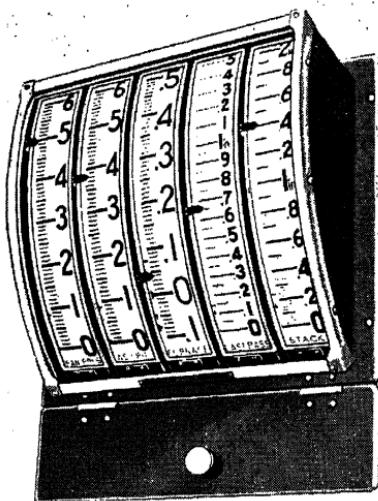


FIG. 181.—Hays multiple draft indicator panel.

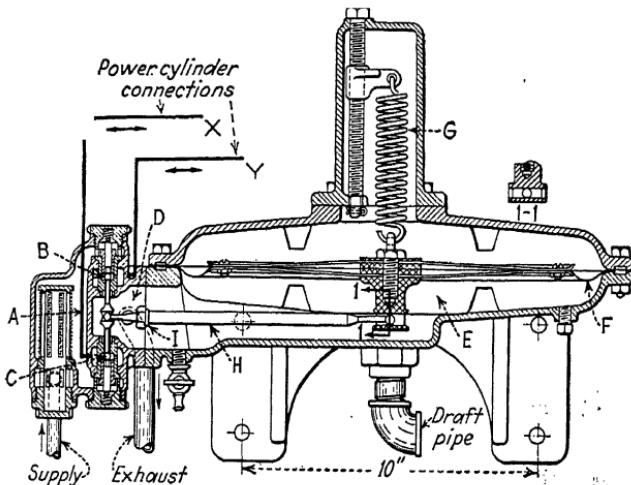


FIG. 182a.—Cross section of furnace-draft controller. Furnace draft is communicated through draft pipe into *E*, and acts upon diaphragm *F* against adjustable spring *G*. Motion of *F* resulting from draft changes is transmitted by lever *H* and diaphragm *I* to valves *B* and *C*, through which fluid flows either to one side of power cylinder or from other side. Since each valve is double-acting, *B* opens its admission port while closing its exhaust port and *C* does the reverse, or vice versa. Hence a change in furnace draft causes the power cylinder to swing the uptake damper enough to restore draft to the point for which the controller is adjusted. (*Courtesy of A. W. Cash Company.*)

produce no accompanying variation in the air damper settings. Hence it is imperative to exercise as close control as possible over oil pressure and viscosity.

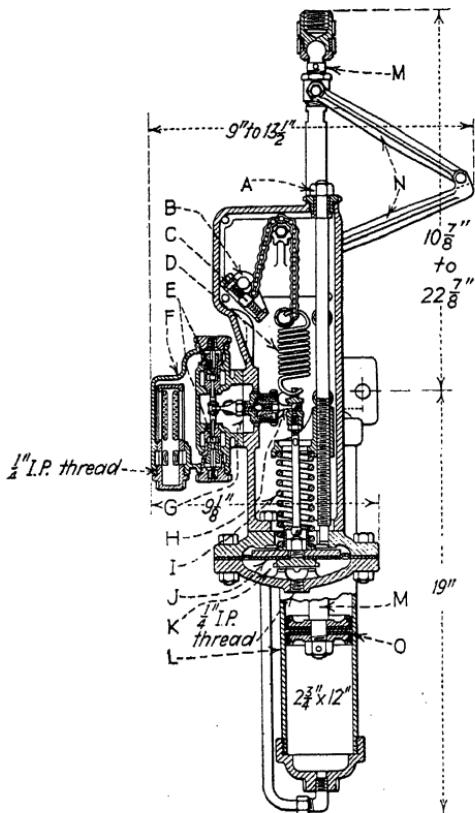


FIG. 182b.—Cross section of steam-pressure controller, hydraulically operated. A change in boiler pressure causes movement of diaphragm *J*. Lever *H* transmits this motion to four-way valve *F* which selectively energizes the correct side of power cylinder *O* and simultaneously exhausts the other side, so that piston *M* moves to operate the apparatus under control in a manner which will restore pressure to desired point. At the same time, through link and lever *N*, compensating lever *B*, and roller chain, this piston movement is transmitted to the compensating spring *D* to change its tension. This alters the effective spring load on control diaphragm *J* in accordance with the piston movement and brings piston rod to rest at proper position. (*Courtesy of A. W. Cash Company.*)

The customary method of pressure regulation is through an oil-pressure relief valve placed at the pump discharge, in the case of electric-driven oil pumps, and steam governor for steam-driven pumps. Uniform viscosity can best be maintained by holding the oil temperature constant, as the viscosity of oil is a

function of its temperature. Viscosity compensating devices are now available only for single-burner installations; as yet nothing has been developed which will automatically compensate for viscosity changes in multiple-burner operation, unless a separate pump be used for each burner.

The plant operator can assist in the securing of uniform oil feed by trying to use oil of fairly uniform specifications and

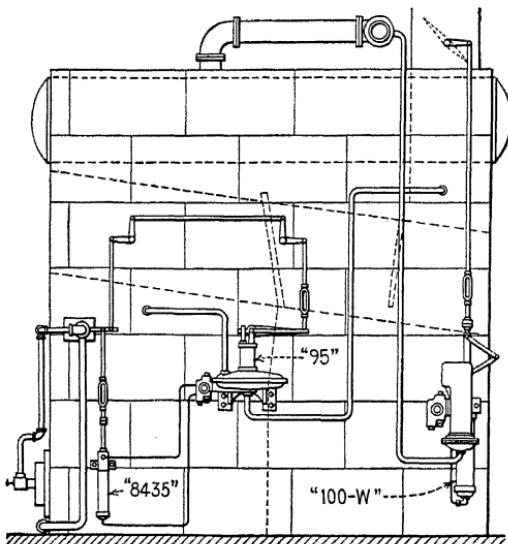


FIG. 182c.—Diagram of complete system of combustion control for natural-draft oil firing of boiler. Steam-boiler pressure is applied to diaphragm of the 100-W controller (Fig. 182b). Movement of that diaphragm causes rise or fall of the piston which is rigidly connected to the uptake damper, bringing about changes in the air flow through the boiler. Loss in draft between boiler furnace and last pass actuates the 95 differential controller (Fig. 182a), which in turn directly regulates the rate of oil feed through the 8435 power cylinder which has its piston connected to control oil valve, primary air, and, if necessary, atomizing medium. (Courtesy of A. W. Cash Company.)

characteristics. When successive deliveries of fuel vary in physical attributes, it may be found that duplicate oil temperatures will not produce duplicate oil viscosity. It then becomes necessary for the operator to study his carbon dioxide records in order that adjustment may be made in the control apparatus to overcome this factor.

5. Compensation for Changes in Pressure Drop through Boiler. So far, the apparatus discussed has been designed primarily to maintain predetermined rates of flow and draft intensities.

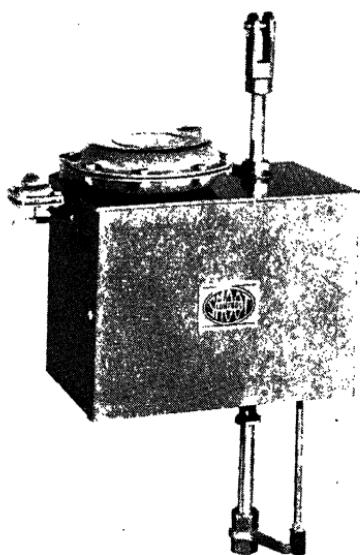


FIG. 183.—Smoot air-flow regulator. (*Courtesy of Republic Flow Company.*)

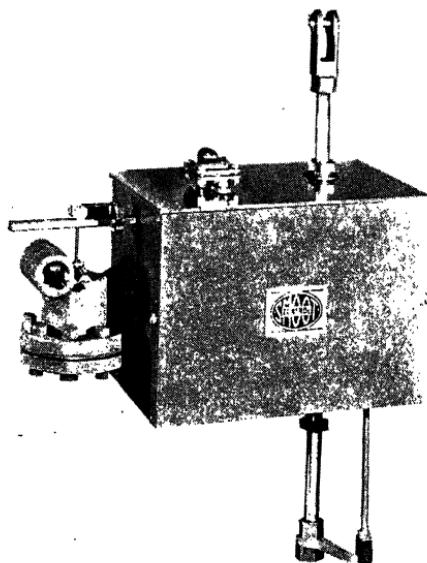


FIG. 184.—Smoot oil-flow regulator. (*Courtesy of Republic Flow Meters Company.*)

So long as ideal conditions exist, such as clean boiler tubes and fairly uniform load, or, at least, fairly uniform pressure drop through the boiler at various ratings, the plant's efficiency can be held satisfactorily high.

But in many cases there exists another factor which cannot be predetermined, for it depends entirely upon instantaneous conditions; this of course is the amount by which the draft is affected by the additional friction caused by soot on the tubes and by the increased velocity of gases through the boiler at high ratings. To meet this situation there has been developed the metering type of combustion controller. This equipment compensates for changes in draft loss through the boiler and constantly corrects the air and oil admitted and the gases discharged to the breeching by maintaining at all times a definite ratio between the draft loss in the boiler and the pressure drop across an orifice plate in the oil feed line. This control also compensates the linkage directly from the carbon dioxide analysis. Thus all the inherent variables can be controlled from one master controller, thereby assuring constant steam pressure and continuous high combustion efficiency.

Briefly, a metering-type controller consists of a pressure sensitive element connected to the main steam header. From this device impulses are transmitted to air- and oil-flow controllers. The air-flow controller is also connected to both the inlet to the first pass and exit from the last pass, so that the differential pres-

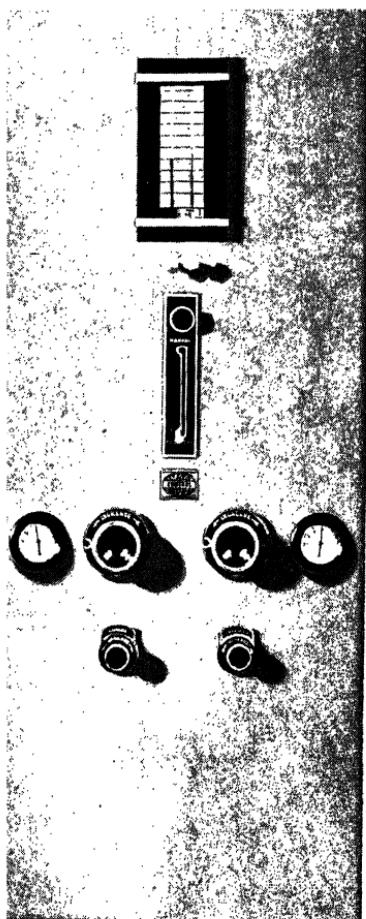


FIG. 185a.—Smoot master controller panel board. (*Courtesy of Republic Flow Meters Company.*)

sure between these points will modify the action of the damper controller in swinging the uptake damper. The oil-flow controller, meanwhile, is also connected to both sides of the orifice in the oil feed pipe, so that the oil-valve opening will be influenced both by the master controller which is responding to changes in steam pressure and by the rate of oil flow as reflected by the

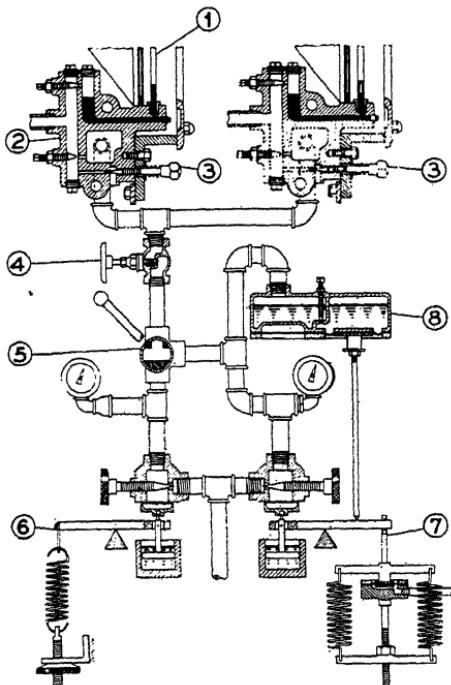


FIG. 185b.—Schematic arrangement of Smoot master controller. 1, manometer; 2, regulator loading element; 3, fuel-air ratio valve; 4, rating ratio valve; 5, transfer valve; 6, manual-control element; 7, steam-pressure element; 8, compensator.

orifice plate. In this manner the oil pressure can be varied to correspond to boiler requirements. An additional reversible motor operates from the carbon dioxide recorder, compensating the linkage for excess air as indicated by each successive gas analysis. For a battery of boilers, only one master controller is necessary, but each boiler must be equipped with individual oil- and air-flow controllers.

The control systems as outlined up to and including Point 4, when properly installed, should give long and effective service,

and are particularly adapted to moderate-sized plants ranging from 100 to 400 hp. Under such conditions the equipment cost represented in the control system will be amply justified by the resultant fuel economy. The metering-control system, outlined in Point 5, is much more elaborate and expensive, and necessarily is restricted to use in very large industrial plants. But when

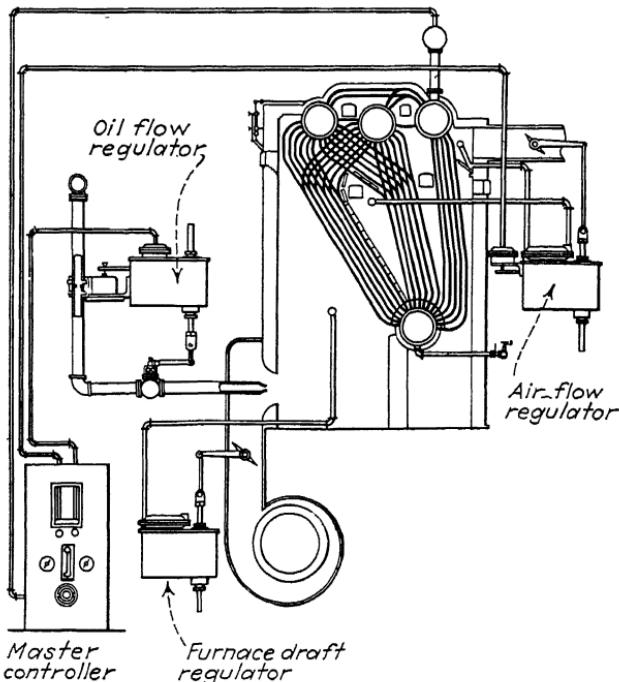


FIG. 186.—Diagram of Smoot boiler-control system.

size of the plant justifies its use, it will soon warrant the investment cost by the resultant high fuel economy of operation.

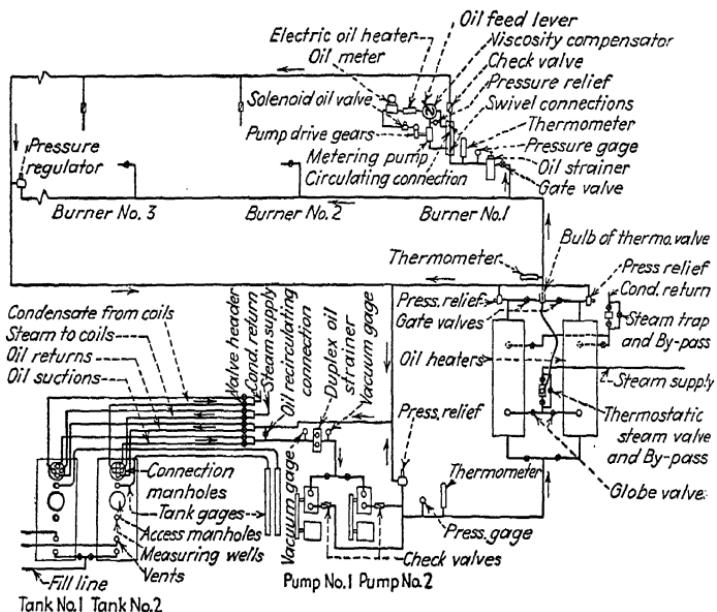
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CHAPTER XVI

PREHEATING FUEL OIL

The viscosity of a fuel oil is an inverse function of its temperature, the viscosity falling as the temperature rises. This fact is of great importance, for were it not so, the great bulk of industrial fuel oils would be unavailable as fuel in liquid-fuel burners. Oil burners and accessory equipment can handle fuel oil only when its viscosity is below that value where it begins to lose its liquid



TYPICAL LAYOUT OF OIL FLOW SYSTEM

FIG. 187.—Diagram of oil-handling system: storage, piping, heating, distribution.

properties and approach those of a solid. Viscosity must, therefore, be controlled to prevent its exceeding that maximum, or it must be reduced to that maximum limit, or even lower, whenever required. The only means of so controlling viscosity is temperature; hence industrial fuel oil must be so handled that the temperature can be varied to suit conditions of viscosity requirements. The general practice of controlling fuel-oil temperature is called preheating.

The principal need for preheating, then, is to bring the viscosity of very heavy fuels within the range of utility as liquid fuel. But there are other factors of lesser importance which also enter, and which make preheating desirable in many cases for other reasons as well. First, since combustion is a chemical reaction, and since all chemical processes are accelerated by increase of the temperature of the reacting substances, preheating of fuel oil results in

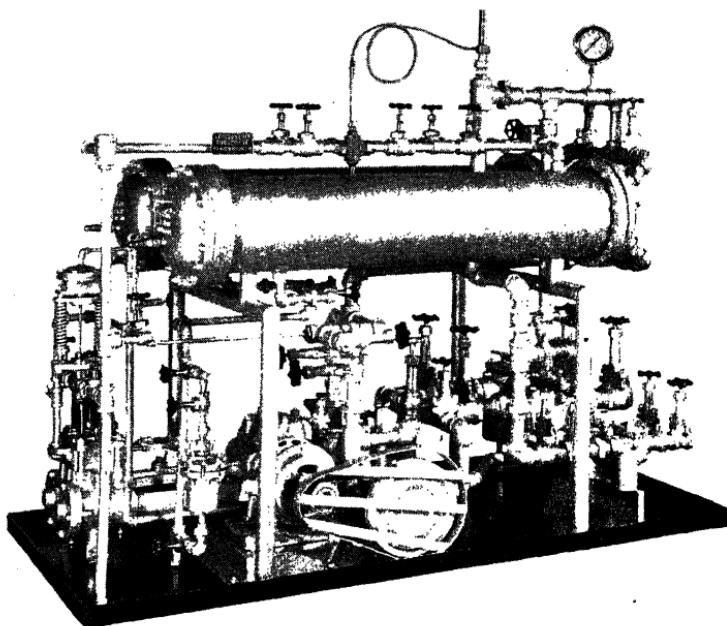


FIG. 188.—Oil heating and pumping unit, using both steam-engine motor drive.

faster and therefore better combustion.¹ Secondly, atomization is also a function of viscosity, and most burners can break up the oil more effectively at lower than at higher viscosities. It follows therefore that better atomization will result from preheated oil than from cold oil, assuming that the cold oil itself could be atomized at all, as would be the case, for instance, with No. 5 oil in an horizontal-rotary burner. Better atomization in turn

¹ This fact would also make desirable the preheating of the air for combustion, but generally speaking, this is not so feasible as heating the oil and is seldom resorted to except in large central-station power plants.

means more intimate mixing with air—hence less air is required. The effect of excess air on efficiency has already been discussed in Chap. III. Finally, as is pointed out in Chap. XII, viscosity affects fluid flow, and for the same number of gallons moved, lower viscosity results in less wear on the pump, less power required for pumping and smaller pipe sizes required for transporting the oil.

The relationship existing between viscosity and temperature is best illustrated by plotting a curve with those two factors as coördinates. Each oil will have its own characteristic curve, the curves taking the general appearance shown in Fig. 190 when ordinary scales are used to represent the temperature and viscosity. These curves indicate a logarithmic relationship from their general form, and this fact of logarithmic relationship is established when the curve is plotted on special coördinate paper (logarithmic). The curve then comes out as a straight line. For interpolation of a curve, to show viscosity at any temperature, the logarithmic coördinates are best, for a straight line can be established from two points and the entire curve can be constructed from data on just two sets of temperature-viscosity determinations. Typical curves on such coördinates are illustrated in Fig. 191.

In Chaps. VI, VII and VIII the matter of maximum viscosity allowable was discussed for each of the different types of oil burners. As a matter of fact, preheating is not generally used on domestic burners, so it is not important to consider variation of temperature and viscosity on the lighter grades of fuel. Many commercial burners use No. 5 oil cold; others find it necessary or desirable to burn it hot. A great many industrial burners employ heated oil, either No. 5 or 6. So for practical purposes, it is sufficient to consider in detail only the heating of the two heavy grades of fuel oil—Nos. 5 and 6.

In general, heating of fuel oil is accomplished by pumping it through a vessel or container which is maintained at a high temperature through an agency called the heating medium. In this vessel the temperature of the oil is increased by absorption of heat from the heating medium. Three heating media are in general use: steam, hot water and electricity. A preheater must therefore embrace these elements; a heating medium, a surface of contact to transfer heat from the heating medium to the oil

and means of conducting both the oil and the medium through the heater. The most important of these is the surface of contact, and to understand the significance of contact surface, it is necessary to consider first the mechanism of heat transfer, or the flow of heat.

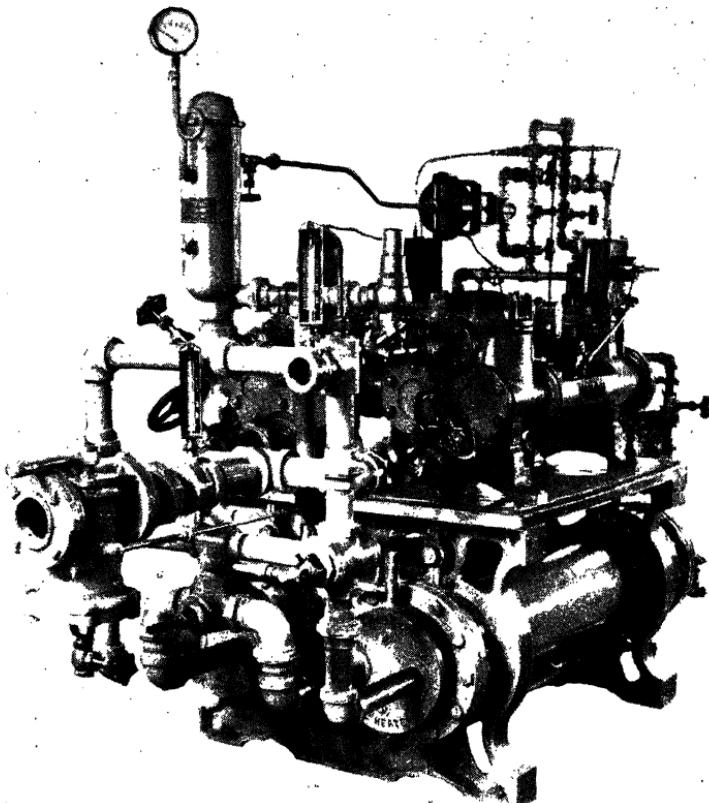


FIG. 189.—Oil heating and pumping unit, with steam drive. (*Courtesy of A. N. Best Engineering Company.*)

THEORY OF HEAT FLOW

In many respects the flow of heat is similar to the flow of electricity. Electric current flows from a point of high potential to a point of low potential, and differences of potential are measured in volts. Heat flows from a point of high intensity to a point of low intensity, and differences of intensity are measured in degrees temperature. Quantities of electricity flowing are measured in

TABLE XLII.—THERMAL CONDUCTIVITIES OF COMMON SUBSTANCES
 $K = \text{B.t.u.}/(\text{hr.})(\text{sq. ft.})(^{\circ}\text{F.})(\text{ft.})$

A. Metals

Substance	Temp., °F.	Source
Aluminum...	64	Marks
Aluminum...	212	Marks
Antimony....	32	10.6
Bismuth.....	64	4.7
Cadmium....	64	53.7
Copper.....	64	222
Copper.....	212	220
Gold.....	64	189
Iron, pure....	64	39
Iron, wrought	64	34.9
Iron, cast....	129	27.6
Steel, 1% C..	64	26.2
Steel, 1% C..	212	25.9
Lead.....	64	20.1
Mercury.....	32	3.6
Nickel.....	64	34.4
Platinum.....	64	40.2
Silver.....	64	244
Tin.....	64	35
Zinc.....	64	64

B. Alloys

Brass, yellow.....	32	*49.4	Marks
Brass, red.....	32	59.5	Marks
Constantan (60Cu, 40Ni).....	64	13.1	Marks
Nickel silver.....	32	16.9	Marks
Manganin (84Cu, 4Ni, 12Mn).....	212	15.2	Marks
Chromium alloy (Cr 13.22, Ni 0.23, Si 0.86, Mn 0.31, C 0.10).....	626	19.3	Martin

C. Liquids

Benzene.....	86	0.0888	Perry
Ethyl alcohol, 70 %....	68	0.138	Perry
Glycerin, (U.S.P. 95 %)	68	0.165	Perry
Heptane.....	39	0.0815	Perry
Hexane.....	86	0.0794	Perry
Kerosene.....	68	0.086	Perry
Petroleum oil.....	59	0.08	Perry
Water.....	32	0.337	Perry
Water.....	140	0.385	Perry

TABLE XLII.—THERMAL CONDUCTIVITIES OF COMMON SUBSTANCES.—(Continued)

D. Gases

Substance	Temp., °F.	K	Source
Ammonia.....	212	0.0172	Marks
Carbon dioxide...	32	0.00791	Marks
Carbon dioxide...	212	0.0122	Marks
Carbon monoxide	32	0.0124	Marks
Ethane.....	32	0.0104	Marks
Ethylene.....	32	0.00947	Marks
Hydrogen.....	32	0.0918	Marks
Hydrogen.....	212	0.0895	Marks
Methane.....	32	0.0170	Marks
Nitrogen.....	32	0.00181	Marks
Oxygen.....	32	0.0134	Marks

E. Miscellaneous Solid Substances

Substance	Apparent density, lb. per cu. ft.	Temp., at room temp.	Source	
Asbestos boards.....	120	68	Perry	
Asbestos sheets.....	55.5	124	Perry	
Asbestos slate.....	112	32	Perry	
Asbestos.....	36	122	Perry	
Asphalt.....	132	68	Perry	
Bricks:				
Alumina (92-99 % Al ₂ O ₃).....		1737	Perry	
Building brickwork.....		68	Perry	
Chrome brick (32 % Cr ₂ O ₃)...	200	392	Perry	
Fire-clay brick.....		932	Perry	
Magnesite (86.8MgO, 6.3 Fe ₂ (OH) ₃ CaO, 2.6SiO ₂).....	158	400	Perry	
Magnesite.....	158	1200	Perry	
Magnesite.....	158	2200	Perry	
Silica (93SiO ₂).....	105	932	Perry	
Silicon carbide (carborundum).	129	2200	Perry	
Concrete (1:4 dry).....			0.44	Perry
Cork board.....	10	86	0.025	Perry
Felt, wool.....	38		0.5 -0.75	Perry
Glass, window.....			0.3 -0.61	Perry
Ice.....	57.5	32	1.3	Perry
Leather.....	62.4		0.092	Perry
Paper.....			0.075	Perry
Paraffin wax.....			0.15	Perry
Rubber, soft.....		70	0.075-0.092	Perry
Sand, dry.....	94.6	68	0.19	Perry
Wood, oak, across grain.	51.5		0.12	Perry

NOTE: Reference to Marks is "Mechanical Engineers' Handbook."

Perry is "Chemical Engineers' Handbook," which see for longer list of values on building, insulating materials and refractories.

terms of amperes; quantities of heat flowing are measured in terms of British thermal units (B.t.u.'s). Conduits of electrical flow are called wires or conductors; conduits of heat flow are called conductors. Amount of electricity transferred in any particular case is determined by the potential difference (voltage) and resistance of the conductor. Amount of heat transferred

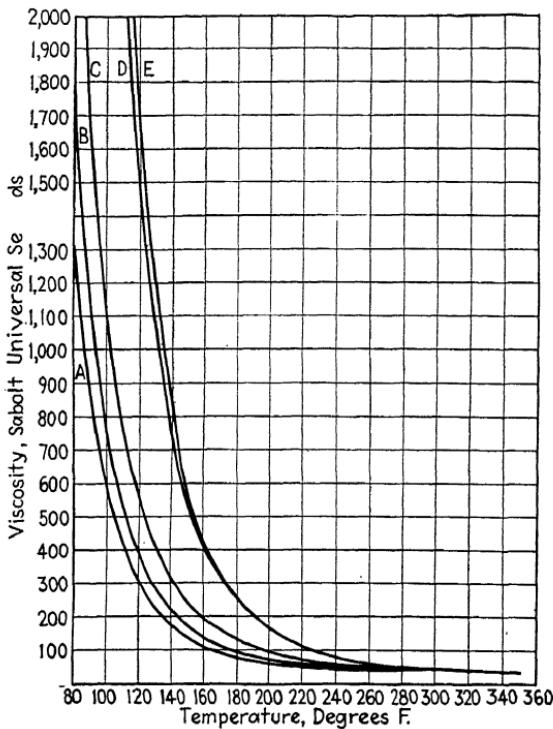


FIG. 190.—Viscosity-temperature chart and typical curves on arithmetic coordinates, for five samples of heavy fuel oil described in Fig. 191.

in any particular case is determined by the potential difference (temperature difference) and resistance of the conductor. The quantity of electricity transferred by a conductor of unit cross section and unit length in unit time, when the potential difference is unity (1 volt), is called the specific conductivity of the conductor, and its reciprocal is termed specific resistance. The quantity of heat transferred by a conductor of unit cross-sectional area and unit thickness in unit time, when the temperature difference is 1° , is called the specific conductivity of the conductor and its reciprocal is termed specific resistance.

For heat, in engineering practice, the unit of area employed is the square foot, the unit of thickness is the foot, the unit time is 1 hr., the degree is on the Fahrenheit scale. In this book, specific conductivity will be called the coefficient of heat transfer and will be measured in terms of B.t.u. transmitted per hour per square foot of area per degree difference of temperature. The coefficients of heat transfer for some common materials are given in Table XLII, page 317.

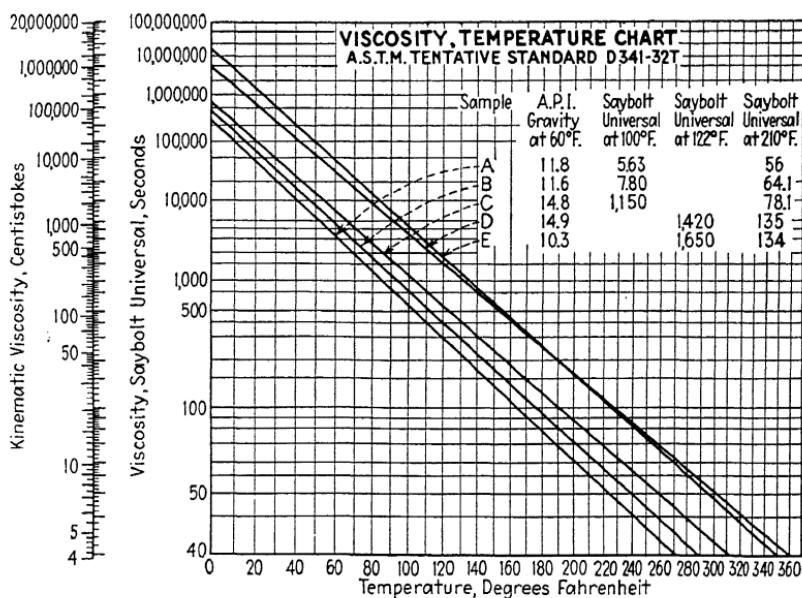


FIG. 191.—Viscosity-temperature chart and typical curves on logarithmic coordinates. (*Chart copyright American Society for Testing Materials, data by Phoenix Chemical Laboratory, Inc.*)

These theoretical coefficients of heat transfer represent the rate of heat transfer through materials under the specific condition of a fixed temperature at each end of the system. Actually, such a condition is seldom encountered in practice, for the instant heat is transferred into the oil which is being heated, the temperature difference changes, and if the heating medium is hot water, its temperature also changes as soon as it imparts some of its heat to the contact surface. Any attempt to work out a system for computing the variation of rate of transfer over a given range or to establish the instantaneous value of the rate of heat transfer,

would be too difficult or tedious for commercial application. Instead, a method is employed which deals with average values of the several factors involved, as will be seen from the following discussion.

The first essential is to establish a means of determining an average value of the temperature difference between the steam and oil, or, in the case of heating oil with hot water as the heating medium, of the average temperature difference between the oil and water.

The second step is to determine the amount of heat to be transferred. This is very simple, for it merely involves the total weight of oil heated in unit time (1 hr.), the temperature rise imparted to the oil and the specific heat (or number of B.t.u.'s to heat 1 lb. through 1°).

Thirdly, another expression is worked out showing in another manner the amount of heat transferred. This is based upon the mean coefficient of conductivity, the area through which heat can flow from the medium to the oil and the average temperature difference. Obviously, the heat so expressed is equal to the heat implied in the preceding paragraph. An equation can thus be set up, with the following factors:

Let P = lb. of oil heated per hr.

W = density of oil, lb. per gal.

G = gal. of oil heated per hr.

t_1 = initial temperature of oil, °F.

t_2 = final temperature of oil, °F.

C = specific heat of oil, B.t.u. per °F. (between 0.4 and 0.5).

A = area of heating surface, sq. ft.

d = logarithmic-mean temperature difference, °F.

K = heat transfer coefficient, B.t.u. per sq. ft. per hr. per degree of temperature difference.

t_s = temperature of steam.

t_{w1} = initial temperature of hot water.

t_{w2} = final temperature of hot water.

As stated above, one expression for the heat transferred is based upon oil quantity, temperature rise and specific heat. Mathematically, the heat absorbed by the oil will be $P(t_2 - t_1)C$. Similarly, the heat conveyed through the heating surface will be $A \times d \times K$. Equating,

$$PC(t_2 - t_1) = AdK \quad (1)$$

To include a statement involving gallons of oil per hour, it is necessary to introduce the density of the oil in pounds per gallon. Calling this W , the equation becomes

$$WGC(t_2 - t_1) = AdK \quad (2)$$

From either of these equations, any one of the variables may be computed if *all* the others are known. It will be shown later that the most frequent use of the formula is to enable a calculation for A , the heating surface. But in order to make this possible, it is prerequisite that more information be available on d and K .

If an arithmetic mean were taken between the temperature of the medium and the average temperature of the oil $(t_2 + t_1)/2$, it would be found that the resulting quantity of heat if figured upon this differential would be at variance with actual results—the amount of the deviation becoming greater as the spread between the steam temperature and final oil temperature became less. It has been shown that the true temperature difference to be used in heat-flow problems is the logarithmic temperature difference, which is expressed as

$$d = \frac{t_2 - t_1}{2.3 \log \frac{t_s - t_2}{t_s - t_1}} \quad (3)$$

the logarithm as here shown being to the base 10; the factor 2.3 changes it to the base e (natural logarithms).

When hot water is the heating medium, its temperature does not remain constant as does that of steam, but instead steadily diminishes from the instant of entrance into the heater until the moment of exit. The equation for average temperature difference then becomes

$$d = \frac{(t_{w1} - t_2) - (t_{w2} - t_1)}{2.3 \log \frac{t_{w1} - t_2}{t_{w1} - t_1}} \quad (4)$$

in which t_{w1} and t_{w2} are, respectively, the initial and final water temperatures. Should the values of the variables in any particular case yield a negative numerator, *i.e.*, $t_{w1} - t_2$ being less than $t_{w2} - t_1$, then these two terms are simply reversed in both numerator and denominator of Eq. (4).

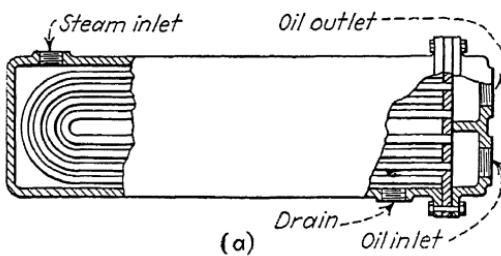
Reverting back to Eq. (1), it is now seen that the only term which still needs amplification is the coefficient of heat transfer, K . The actual value of K depends upon numerous factors. It is an over-all coefficient, representing the sum of the coefficients of the series of transfers which actually compose the system; steam to film on the steam side, film to metal, through the metal, metal to film on the oil side and film to oil. One way of making an actual determination of K would, of course, be to experimentally measure all the quantities involved (temperatures, weights, surface, etc.), substituting these in Eq. (1) and solving for K . But such an experimentally determined value would be correct only for the specific conditions under which the experiment was conducted. For instance, even if all the quantities so far mentioned were to be held constant, but the velocity of the oil through the heater were varied, it would be found that an entirely different value of K would prevail. Even if the velocity were also held constant, and another determination made on a different grade of oil, again K would change.

If all other conditions of the heat transfer could be held constant, the relationship between K and the velocity, V , is expressed by $K = c\sqrt{V}$, where c is a constant for each separate grade of oil. V in this use is in feet per second, and is numerically obtained by dividing the cross section of the path of flow, in square feet, into the flow expressed as cubic feet per second. Just what limiting values should be used for V is a matter of practical experience; it cannot be too high as that would create unnecessary friction losses, and cannot be too low as that would cause deposition of sludge on the walls of the heater tubing.

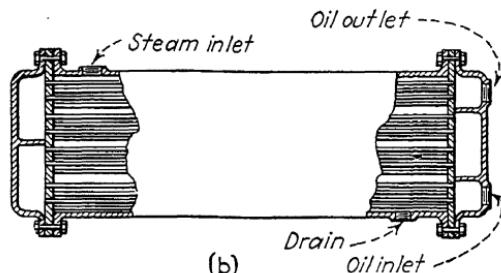
All of the foregoing discussion of K shows that a practical value for it cannot be theoretically determined. Manufacturers of heat-transfer equipment design their equipment upon K values which they themselves have determined from their own experience and experiment. Definite values of K for any specific application cannot safely be given here, because of the complications which have already been pointed out. Mature experience must be available in order to choose a proper and dependable value of K for the individual conditions existing under any given set of circumstances. It is not the intent of this discussion to set forth a method for designing an oil heater; rather it is the hope that this discussion, by indicating the technical problems involved, will

lead to consultation with a competent authority whose judgment will be modified by actual experience under repeated analogous situations and proved performance of equipment of a similar nature.¹

The experienced designer, having selected the proper K value for a specific job and having worked out a suitable value for d , inserts these terms, together with the other required, in Eq. (1) and then solves for the required heating surface. Once the



(a)



(b)

FIG. 192.—Straight-tube heaters; (a) two-pass; (b) four-pass.

amount of heating surface is known, then it becomes a matter of deciding upon a style of construction.

TYPES OF OIL HEATERS

Commercial forms of oil preheaters can be classified as follows:

1. Shell and tube type with heating medium in shell.
 - a. Straight tube.
 - b. U tube.
 - c. Spiral coil.
2. Shell and tube type with heating medium in tube.
3. Concentric tubes.

¹ Some commercially used values of K will be found in the Appendix.

While, of course, each of these types has its advantages and disadvantages, some of them have in their favor only the element of low first cost. Others may involve a somewhat higher investment, but offset this by better performance in operation. Some of the factors to consider in judging the merits of any one type of heater are: ease and cost of installation, accessibility for cleaning, simplicity of parts replacement, compactness in form and space requirements, etc.

The straight-tube type seems to be a popular form of oil pre-heater, and most manufacturers offer some style of straight-tube heater, some also making other styles. One of its manifest points of advantage is that the tubes can be cleaned without the necessity of opening any pipe connections. Generally speaking, these heaters are built with steel shells and steel tubes; the steel shell offers great strength for resisting high working pressures and the steel tubes offering better resistance than would copper tubes to the corrosive action of the sulphur which is present, in varying amount, in all fuel oils.

In selecting an oil heater for a specific application, the following should be duly weighed, for they control the degree of satisfaction which will be derived from the heater's operation:

1. Proper working and test pressure. All high pressure vessels should be built in strict conformance to A.S.M.E. requirements.
2. Number of tubes per pass and the diameter of the tubes, for these two determine the velocity.
3. Number of passes, which, together with No. 2 above, determine the heating-surface area.
4. The actual amount of heating surface which should be certified by the builder as adequate under the conditions of service.
5. Diameter of the tubes, which must not be so small as to offer difficulty in cleaning and unduly increase the friction losses, and must not be so large as to permit stratification within the tube.
6. Capacity must be ample, and should be sized for the total oil pumped, so that surplus oil may be returned hot to storage.

Every installation of an oil preheater should include a pressure-relief valve to guard against excessive pressures resulting from oil expansion.

Automatic temperature control should be provided for maintaining the oil within the desired limits of temperature. When steam is the heating medium, this is readily accomplished by throttling the steam with a thermostatic steam valve actuated from a bulb immersed in the stream of hot oil. When hot water is the medium, thermostatic control is not so easy, although if the water is pumped through the heater by a circulating pump, the pump motor can of course be placed under thermostatic control. Otherwise, thermostatic valves in the gravity water-circulating line can also be used, but will not give the close control which is possible in the case of steam.

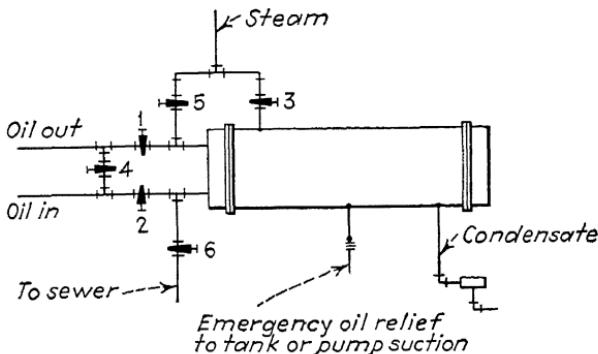


FIG. 193.—Method for cleaning oil heater without disconnecting piping. By closing valves 1, 2 and 3, and opening valves 4, 5 and 6, steam is sent through the heater tubes to clean them.

Proper pipe size for steam to serve the heater can be calculated from knowledge of the weight of steam required, which in turn is derived from the total B.t.u. absorbed by the oil. Reference to standard steam-flow tables will then reveal the correct pipe size. The following formula can be used to compute the weight of steam needed for preheating:

$$\frac{P(t_2 - t_1)c}{970} = \text{lb. of steam per hr.} \quad (5)$$

A steam trap of ample capacity to handle the condensation should be installed to avoid water hammer and for economy of steam consumption. A bucket trap is the preferred form for this use.

Hot-water exchange-type heaters should be tapped into the boiler about 4 in. below the normal water line. The pipe from

the boiler into top of heater should pitch toward the heater to facilitate circulation, and for the same reason, the return pipe from heater to boiler should pitch toward the boiler. Circulating pumps on larger sizes of heaters avoid the need for running large pipe sizes, and increase heater capacity somewhat by increasing the velocity of the water.

ELECTRIC OIL HEATERS

The theory of electric oil heaters is relatively simple. The mathematical relationship between electricity and heat is that 1 kw.-hr. of electrical energy is equivalent to 3415 B.t.u. From this it is easy to set up a formula for calculating the required

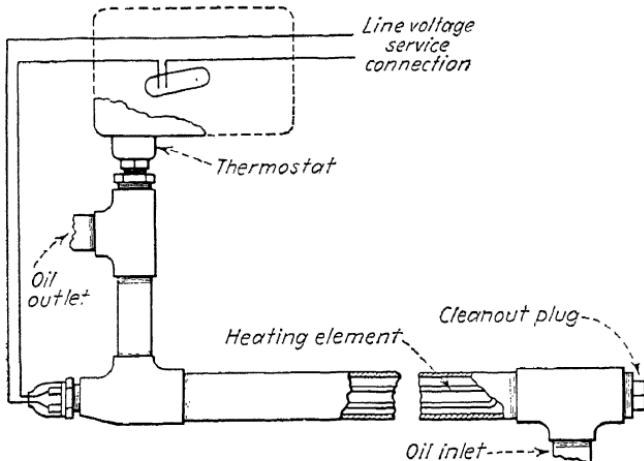


FIG. 194.—Electric oil heater.

heater capacity to accomplish any desired heating task. The desired quantity of oil to be heated should be expressed in pounds; this when multiplied by the degrees of temperature rise to be imparted to the oil, and again multiplied by the specific heat of the oil (0.5), will give the total heat input per hour in B.t.u. Dividing the total number of B.t.u.'s needed per hour by 3415 gives the kilowatts required. In heating by electricity, the efficiency of heat conversion is almost 100 per cent, so that it is not necessary to allow any excess heater capacity to compensate for losses.

Several manufacturers offer excellent types of ready-made electric heater elements for this work which can be directly incorporated into oil heaters. The heater itself consists essentially

of a shell containing the oil, the heater elements and a thermostatic regulator to control the current from the oil temperature. When the electrical load is fairly high, above 1000 watts, it is necessary to use either a high capacity thermostat to handle the heater current, or else a relay to handle the heater current while the thermostat controls the circuit which actuates the relay.

The design of the heater should always permit of quickly removing the electrical element, should it require replacement, without the necessity of disturbing the oil connections. The passage of oil through the heater and over the heating element must be so laid out that short-circuiting of oil through the heater cannot occur, but the oil must be directed so that it comes into contact with the heating surface of the electrical element as long as possible.

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CHAPTER XVII

OIL STORAGE TANKS AND GAUGES

Metallic oil storage tanks are constructed of hot-rolled, low-carbon steel plate, commonly referred to as tank plate. The ultimate failing stress in either tension or compression is assumed at about 55,000 lb. per sq. in., and it is customary to allow a factor of safety of 3 for low-pressure vessels and of 4 for high-pressure vessels. So for average oil storage tanks which are designed for a maximum working pressure of 5 lb. per sq. in. the working strength used is around 18,300 lb. per sq. in. In the case of special tanks designed for higher working pressures or even, when specified, for higher test pressure, the working strength is reduced to 14,000 lb. per sq. in. Tanks built for and tested at 5 lb. pressure are labeled as Underwriter Approved tanks. Tanks built for and intended for test or use above 5 lb. must be constructed according to A.S.M.E. code and regulations.

The plate thickness required in the walls of a cylindrical tank can be calculated from the following formula, in which t is the plate thickness in inches, P is the internal pressure in pounds per square inch, D is the internal diameter in inches and s is the allowable working stress in pounds per square inch:

$$t = \frac{PD}{2s}$$

The computation of stresses set up in the ends of an horizontal cylindrical tank are somewhat more complex, and ordinarily these tanks are not designed from fundamental steel structural formulas. There are by now well-established plate thicknesses for use in standard sizes of oil storage tanks, and the National Board of Fire Underwriters sets up data also, which generally coincide with usual practice. Table XLIII (page 337) gives the National Board data.

UNDERGROUND TANKS

The National Board makes the following recommendations on underground oil storage tanks:

1. Whenever and wherever possible or feasible, oil storage tanks should preferably be buried outside of buildings.
2. Depth should be sufficient so that top of tank is 2 ft. below grade. Backfill should be 1 ft. deep, of earth, and then a concrete slab at least 4 in. thick, extending at least 1 ft. beyond the tank in all directions. If the tank be under basement floor, the concrete slab is mandatory.

Whenever tanks are buried underground, there is a possibility of the tank's settling after the installation is completed and the tanks in use. It is therefore necessary that all pipe connections to the tank be made with swing joints. These permit the tank to move without consequent breaks or leaks occurring in the pipework. The single exception, of course, is the measuring

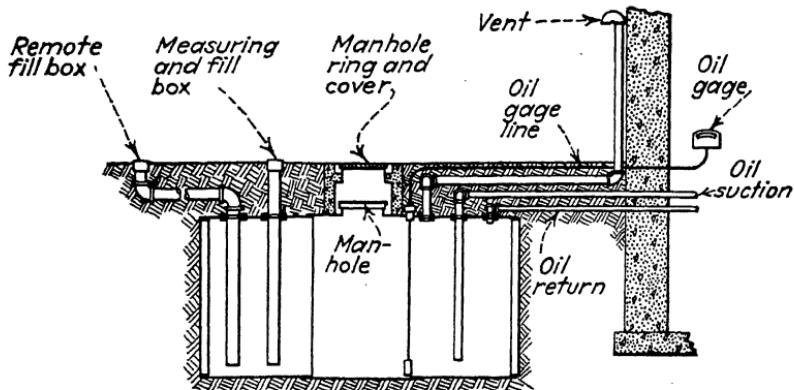


FIG. 195.—Underground oil storage tank and connections.

line, which must come straight up to grade to allow a stick to be dropped into the tank.

A steel tank should never be buried in cinders, or filled land, for such ground contains acids which very quickly corrode the steel. A sandy soil is the best suited to tanks, for sand allows water to drain away rapidly and so keeps the tank as dry as possible. Whenever possible, the backfill should be of sand.

It seems to be a general practice to set underground tanks with one end slightly lower than the other, so that water will collect at the low end, from which point it can be withdrawn through a pipe. But it is doubtful whether such sloping is necessary to facilitate pumping out water, and it is certain that when a tank is not level all readings taken of the tank's contents by depth measurement will be in error. For the sake of accurate oil measurements, it is desirable that the tank be set level. If how-

ever it be desired that the tank have a low end, then the slope should not be greater than $\frac{1}{8}$ in. in 1 ft. When so sloped, it is imperative that the vent line be taken from the high end of the tank.

If a heating coil be used, a helix surrounding the suction stub is to be preferred to a series of straight pipes along the tank bottom. If the oil is returned hot from the pump, the return should enter the tank at a point close to the suction, and the return should be carried down into the tank by a stub reaching to within a few inches of the bottom. Unless facilities are available for raising condensate out of the coil (as a good vacuum pump), it is difficult to secure satisfactory results from a steam coil in a tank, and frequently hot water is circulated through the coil instead. For most commercial installations, better results are obtained by returning hot oil to storage from the burner, than by trying to heat by means of a coil in the tank.

When the conditions of installation are such that there is likelihood of water surrounding the tank, especially when it be empty, provision must be made against the tank floating or being buoyed upward. This is accomplished by anchoring the tank to a concrete slab. The combined weight of the tank and the slab must exceed the weight of the water displaced by the tank. Since the average weight of concrete is 150 lb. per cu. ft., and if V_c is the volume of concrete in cu. yd., V_t is the volume of the tank in gallons, and W_t is the weight of the tank in pounds:

$$V_c = \frac{8.4V - W_t}{4050}$$

Standard cylindrical tanks are practically always welded, and arc welding is to be preferred to acetylene welding. Riveted tanks are seldom used on oil, as the joints cannot readily be made oiltight. Sometimes oil tanks are formed, lightly riveted and then the joints are welded. Since the efficiency of a welded joint is at least 80 per cent, while that of a single rivet seam is seldom above 65 per cent, there does not seem to be much point to combining riveting and welding.

All steel tanks should be painted with two coats of asphaltum or bitumastic paint to resist corrosion. This applies to above ground tanks as well as buried tanks.

A manhole is a very desirable adjunct to any oil tank, so that periodically access may be had to the tank's interior for cleaning. Small tanks, especially when used for light-grade oils, as in domestic installations, do not accumulate much heavy sediment, and an annual pumpout of the settling from the bottom is about all that is required. But whenever heavy oils are being burned a certain amount of sludge settles out of the oil and accumulates on the bottom. Much of this can be removed by suction through a pipe or hose, but sometimes it becomes necessary to open the manhole to send a man in who will have to use a shovel and scrape to really clean the tank properly. Manholes are fitted with flanged covers bolted to the flange on the tank. A masonry well should be built around the manhole, with a cover plate at the grade. The diameter of the manhole should be sufficient to allow a man to reach in and unscrew the bolts on the flanged cover and, especially, to permit reclamping the cover in place by reaching underneath the flange for reinserting the bolts.

The National Board lays down no regulations on underground tanks to govern the location with respect to building foundation walls. But many local rules and ordinances do prescribe a minimum distance between building and tank, and such requirements should be ascertained and obeyed before tank location is decided upon. Most municipalities and other local governments require also that permits be secured for tank installations.

TANKS INSIDE BUILDINGS

The National Board makes the following rules and suggestions on tanks inside buildings.

1. Tanks larger than 60 gal. capacity shall be in the lowest level of the building.
2. Unenclosed tanks must be at least 7 ft. from boiler or fire.
3. Maximum sized tank without enclosure permitted is 275 gal. Maximum aggregate capacity permitted is 550 gal.
4. Tanks exceeding capacity of (3) must be enclosed in either 6 in. of concrete or 8 in. of brick, bonded to floor. The wall shall be carried up to within 1 ft. of the ceiling and the space between the tank, the enclosure and the ceiling shall be filled with sand or earth. If the ceiling is not fire-resistant, a 5-in. slab of concrete shall be placed on top. Or instead of the above, the tank may be completely enclosed within 6 in. of reinforced concrete.
5. In ordinary buildings tanks shall not exceed 15,000 gal. capacity.

6. In fire-resistive buildings the tanks shall not exceed 15,000 gal. capacity.

7. In any building, if the tank room be fire-resistive and isolated from the rest of the building, the tanks shall not exceed 50,000 gal. total capacity, with individual tank capacity not to exceed 25,000 gal.

Inside tanks should also be equipped with manholes, if possible, especially if No. 6 oil is to be used. If heating coils are needed, the same remarks apply as given above on underground

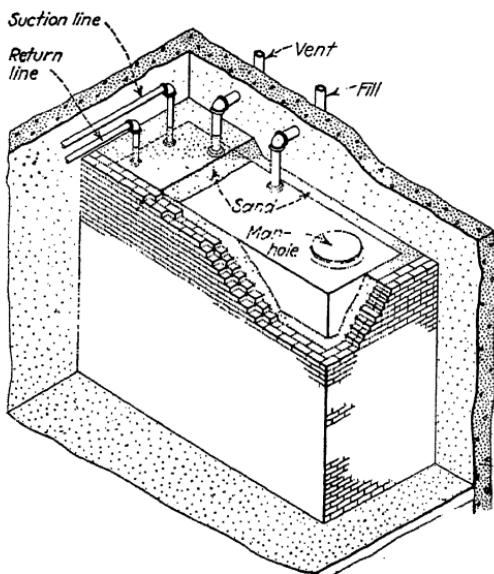


FIG. 196.—Basement oil storage tank and connections, with brick enclosure and sand fill.

tanks. Inside tanks should be carefully leveled off and set on a flat concrete surface.

By far the greater majority of inside tanks are erected on the premises by welding of flat plates. Electric-arc welding gives such superior results over acetylene welding that it should be employed whenever possible. This implies, of course, that a portable apparatus be available for generating the electric current—the customary type consisting of a gasoline engine driving a direct-current generator.

The dimensions and plate thicknesses of standard rectangular tanks are shown in Table XLIV. Tanks up to and including the 1500 gal. capacity do not require bracing, but larger or longer

tanks require braces across the tank to tie together the side walls. For tanks 4 ft. high, braces should be 6 ft. on centers; for 6 ft. high, 4 ft. on centers; and for tanks 8 ft. high, 30 in. on centers. The brace consists of a steel bar welded on each end to angle irons which are welded to the respective side walls at the center of the height.

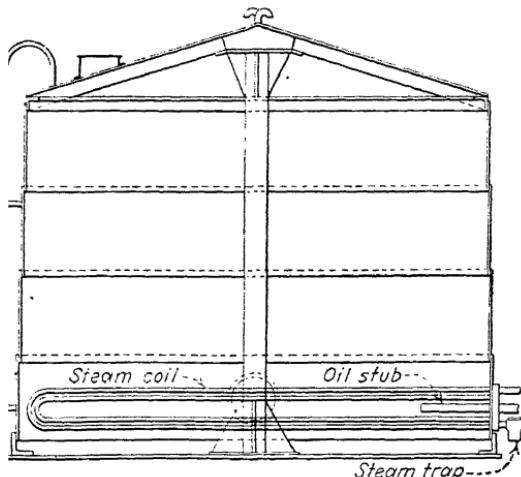


FIG. 197.—Cylindrical vertical tank, aboveground.

ABOVEGROUND TANKS

Aboveground tanks find their chief use in bulk storage plants; consumers of fuel oil seldom use them, but rather install either underground or inside tanks. The National Board sets up the following regulations for aboveground tanks:

1. The minimum distance between tanks and either property line or nearest building shall be:

Capacity, gal.	Distance, ft.	Capacity, gal.	Distance, ft.
750	5	128,000	75
1,100	10	200,000	85
3,000	20	266,000	100
21,000	25	400,000	150
31,000	30	666,000	250
45,000	40	1,333,000	300
64,000	50	2,666,000	350
80,000	60		

2. Over 400,000, the distance may be only 175 ft. if approved fire extinguishing equipment be installed.

3. For tanks permitted from 50 ft. up to 175 ft., capacity may be increased 33 per cent by installing approved fire extinguishing equipment.

4. The minimum distance between tanks shall be:

Capacity, gal.	Distance, ft.	Capacity, gal.	Distance, ft.
18,000 or less	3	75,000 or less	13
24,000 or less	5	100,000 or less	15
48,000 or less	10	2,500,000 or less	1 tank diameter

5. Sites shall be selected that offer little hazard to surrounding property. Where there is likelihood, should the tanks break or be overflowed, of flooding surroundings, suitable dikes or embankments shall be erected around the tanks.

TANK PIPING

Fill pipes, according to the National Board, shall terminate outside of building, at a point at least 5 ft. from any opening in the building at the same or lower level. Tanks shall be filled only through fill pipes, and the fill pipes must be tightly capped when not in use. When tanks are to be filled by pumping, the fill pipe should be at least 2-in. pipe. When tanks are to be filled by gravity, the fill pipe should be 2 in. on light oils and increase as the oils become heavier, up to 6 in. for No. 6.

It is good practice to make the vent pipe at least as large as the fill pipe, up to 2 in. In no event, according to National Board requirements, shall the vent be less than 1½ in. Vent pipes should be arranged to drain toward the tank, and the tank end should not extend into the tank more than 1 in. Vent pipes shall terminate outside of buildings, at a point not less than 2 ft., vertically and horizontally, from any building opening. The outer end shall be provided with a weatherproof hood, and shall be sufficiently high above the ground to clear deposits of ice and snow. No cross connection shall be permitted between the vent pipe and any other pipe. The vent should always terminate at a level higher than the fill.

Suction pipes should always clear the tank bottom by a few inches, to permit settling, including water, to accumulate.

When steel pipe is used, the suction stub is generally screwed into the lower tapping of a double-tapped bushing which is then itself screwed into the tank flange. The line then continues by another pipe screwed into the upper tapping of the bushing. The need for the double tapping is, of course, obvious, for if the pipe thread were cut continuously from one side of the bushing to the other, it would be so deep that the joint between the pipe and bushing would be so loose as to make an airtight seal very difficult. In the case of copper tubing, the tubing is run continuously from the building wall, through the tank opening, down to within a few inches of the bottom, a special fitting being used in the tank flange to hold the tubing rigidly at that point and to make the opening watertight (if the tank is buried).

For many years it has been regular practice to provide foot valves, which are vertical check valves, at the end of the suction stub. The purpose is to keep the suction line filled with oil when the pump stops running. On light oils this foot valve does no special harm, but in the case of heavy oils, the value of a foot valve is dubious. Very frequently the check mechanism becomes stuck when gummy deposits of tar collect around it, and should it stick in the closed position the suction line is, of course, rendered useless. And for that matter, should it stick in the open position it no longer serves its intended purpose. So it seems to be a sensible thing to dispense with foot valves on commercial and industrial oil-burner jobs.

Antisyphon valves, for the most part, can be placed in the same category as foot valves, insofar as practicality is concerned. Recognition of their futility is at last conceded by the National Board, at least partially, for the Board's regulations have been modified to the extent of stating that antisyphon devices are "advisable," instead of "mandatory" as heretofore. Most antisyphon valves operate on the principle of minimum vacuum required to raise the check valve. Usually, this minimum vacuum is about 10 in. When it is realized that very seldom does ordinary operation of the pumping system require more than 3 or 4 in. of vacuum to draw oil over from the tank, it can be appreciated how much additional load is thrown upon the pump by the pressure of the antisyphon valve. This additional vacuum requirement causes so much unnecessary interruption of routine operation that most users sooner or later remove the check device

from the valve. Such being the case, it surely is only logical to recognize the uselessness of this contraption and to abandon its use. The contingency against which it is supposed to be a safeguard—breaking of the suction line in such manner that oil syphons over from the tank onto the floor—is very remote; it is doubtful if many accidents of serious consequences have occurred through such means. Reliable pipe fitting is a much more sensible way of avoiding such hazards.

Underground suction lines should always have two unions, one at the tank top and one near the building wall. Then there is always a fairly easy means of opening the line to remove any obstructions that should occur (referring to steel pipe lines). All underground piping will give longer life if it be galvanized, including the fittings. When the lines must be run through or under cinders, the pipes should be encased in tile conduit, with the joints made watertight. Otherwise corrosion will soon destroy the pipe.

All underground tanks require a measuring line, through which a gauge stick can be inserted to verify the tank contents. The top should have a cap during normal operation, to be removed only when the oil level is measured. The National Board has an unqualified rule against using an opening on an inside tank for stick measurement. This seems to be universally disregarded, for it is certainly general practice always to provide a spare opening in inside tanks for just this purpose. And judging from the total absence of ill effects from this way of measuring tank contents, it seems that the rule is entirely unnecessary.

SELECTION OF TANK SIZE

In determining what capacity tank to use in a particular installation, there are several factors to bear in mind.

1. For heating plants, during the coldest month, usually January, as much as 20 per cent of the entire year's consumption may be burned. The tank should hold, at least, a quantity sufficient for one week of continuous cold weather. For example, if it is known or estimated that a building will burn 40,000 gal. annually, then as much as 8000 gal. may be burned during the coldest month, or 2000 gal. per week. Allowing for the suction-stub clearance above the tank bottom, it is clear that the minimum safe tank capacity would be 2500 gal.

2. For a carload user, there must be ample reserve to allow for delays in car shipments. Also, it should be remembered, most fuel-oil tank cars hold 10,000 gal. Hence, if there is to be some reserve on hand when a tank car is ordered, with the possibility

TABLE XLIII.—DATA ON CYLINDRICAL TANKS FOR UNDERGROUND
INSTALLATION*
A. National Board Requirements

Capacity, gal.	Minimum thickness	Weight, lb. per sq. ft.
7 to 285	16 gauge	2.50
286 to 560	14 gauge	3.125
561 to 1,100	12 gauge	4.375
1,101 to 4,000	7 gauge	7.50
4,001 to 12,000	$\frac{1}{4}$ in. (nominal)	10.00
12,001 to 20,000	$\frac{5}{16}$ in. (nominal)	12.50
20,001 to 30,000	$\frac{3}{8}$ in. (nominal)	15.00

B. Commercial Dimensions

Capacity, gal.	Diameter, in.	Length, in.	Thickness, in.
550	48	72	$\frac{3}{16}$
1000	64	72	$\frac{3}{16}$
1000	48	132	$\frac{3}{16}$
1500	64	108	$\frac{3}{16}$
1500	48	198	$\frac{3}{16}$
2000	64	144	$\frac{3}{16}$
2500	64	180	$\frac{3}{16}$
3000	64	216	$\frac{3}{16}$
4000	64	288	$\frac{3}{16}$
5000	72	288	$\frac{3}{16}$

* Notes: On National Board requirements, if adequate internal bracing is provided, tanks of 12,001 to 30,000 gal. capacity may be built of $\frac{1}{4}$ -in. plate. For tanks larger than 1100 gal. capacity, a tolerance of 10 per cent in capacity is permitted.

that a car delivery may vary from two days to a week, it is apparent that 12,000 gal. is an absolute minimum so than an entire 10,000 gal. can be unloaded without the necessity for reducing the amount on hand below 2000. And 15,000 gal. tank capacity is much safer (speaking now of a moderate user). In large industrial plants where the consumption is really high, storage capacity

must be computed on holding, at least, a full week's supply continuously on hand.

3. Fuel oil being an active commodity on the nation's markets, its price is subject to market fluctuations. Very often it is

TABLE XLIV.—DATA ON RECTANGULAR TANKS FOR BASEMENT STORAGE

Capacity, gal.	Width, in.	Length, in.	Height, in.	Thickness, in.
530	54	54	42	10 gauge
1000	54	89	54	$\frac{3}{16}$
1500	54	120	54	$\frac{3}{16}$
2000	60	132	60	$\frac{3}{16}$
2500	60	134	72	$\frac{3}{16}$
3000	72	132	72	$\frac{3}{16}$
4000	72	180	72	$\frac{3}{16}$
5000	72	226	72	$\frac{3}{16}$

possible to purchase quick offerings of fuel oil at attractive prices. Ample storage makes such advantageous purchasing possible.

Frequently, additional storage repays the extra original cost many times over during its lifetime.

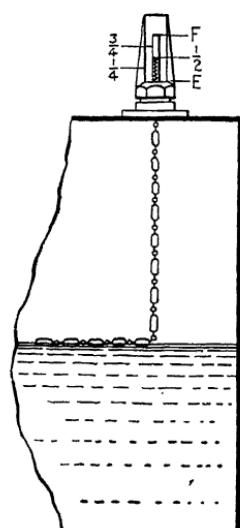


FIG. 198.—Linked float tank gauge.
(Courtesy of Detroit Lubricator Company.)

TANK GAUGES

The use of gauge glasses on inside tanks for indicating tank contents is strictly prohibited by all Underwriter rules. Such apparatus is a decided fire hazard, and under no conditions should it be used.

Approved tank gauges are of two general types, float operated and hydrostatically operated, and there is a third type which employs a combination of the two. Simple float gauges are used on inside tanks for direct reading from the top of tank; hydrostatic and combination float-hydrostatic gauges are used to give indirect readings at point remote from the

Simple float gauges are not accurate even when operating properly and, frequently, do not operate freely or at all. At best,

the float gauge offers a means of judging when the tank supply is low enough to requiring reordering of fuel. It cannot be used as a means of checking amounts of delivery or amounts on hand within any degree of accuracy. Hydrostatic gauges are available that offer a very accurate means of determining a tank's contents; when properly designed and installed they will give correct readings within an accuracy of 1 per cent.

The simplest form of tank gauge is the float type. In most designs the gearing and other parts are very crude, so that close indications of liquid level cannot be expected. In addition, the float, especially in heavy oils, becomes covered with a coating of grease and tar, and loses partially or completely its original buoyancy. Some designs overcome the inherent defects of the gearing by employing a magnet to turn the indicating hand over the dial. In this manner the motion lost in the customary gear arrangements can be minimized or even eliminated to a degree permitting of fair accuracy. Other styles of float gauge simply communicate the rising or falling motion of the float on the liquid surface to a wire or string which in turn passes over a pulley and raises and lowers an indicator over a scale, or the entire pulley may be used as the indicator by having a scale appear on its cylindrical surface. An interesting example of a float gauge is shown in Fig. 198, where the buoyancy effect is translated into weight effect by a continuous linkage of small floats. But, in general, float gauges of all types should be restricted to use on domestic installations or light oils, and, of course, they are all intended and applicable for installation where a reading can be made on the tank top.

HYDROSTATIC GAUGES

The operating principle of the typical hydrostatic gauge is diagrammatically outlined in Fig. 199. In essence, an air chamber which rests on the tank bottom communicates through a tubing to the indicating instrument located at a convenient spot; through suitable apparatus, the pressure variations, which occur to the air trapped in the conduit by reason of the increasing or diminishing hydrostatic head as the liquid level rises and falls, are translated into readings on the scale of the instrument. It is thus seen that this type of gauge employs not only hydrostatic pressure but pneumatic pressure as well. In fact there must

always be provided a means of renewing the air in the conduit, which is sometimes expelled when the tank is overflowed, and which sometimes escapes through leaks in the tubing and its connections. Usually a small air pump is made a permanent part of the apparatus, or else a capped opening is left into the conduit so that air may be pumped in by connecting a pump whenever required. In at least one design, as illustrated, a tubing is also used between the upper end of the U tube of the indicating instrument and the tank itself, so that should the tank be overflowed the excess pressure can force the liquid of the U tube only up to the top, but not over. The conventional instrument consists of the U tube illustrated, and the tank content is read by taking the position of the top of the liquid column in the free leg of the

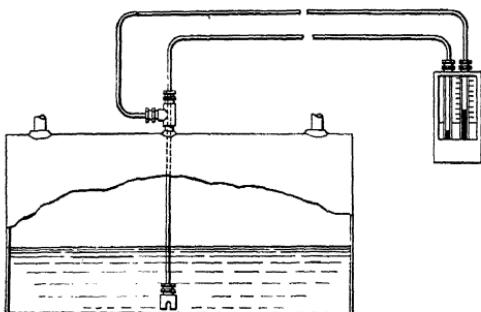


FIG. 199.—Hydrostatic tank gauge. (Hytometer.)

U tube. Other models have an arrangement for translating the air pressure into the movement of a needle over a scale.

All hydrostatic gauges must be calibrated for the specific gravity of the liquid in the tank, which is to say that any given scale on the instrument will be correct for only one grade of fuel oil. One make of hydrostatic gauge is mounted upon an adjustable pivoting arrangement, whereby the inclination of the U tube may be varied to compensate for changes in the specific gravity of the fuel oil being measured. It should be noted that since all liquids change their specific gravity with temperature change, the scale must show the temperature for which it has been calibrated, and any readings taken at a temperature of fuel oil other than the standard must be corrected accordingly.

In the combination float and hydrostatic gauge, the motion of the float on the liquid surface in the tank is utilized to operate a

system of bellows filled with a fluid. These parts are shown in diagram in Fig. 200, as well as the indicating instrument which is operated by the bellows so that a needle moves over a scale to show the height of liquid or contents of the tank. To be dependable in both operation and accuracy of its readings, this gauge must be designed with a good system of gears and bearings so that the motion of the float may be free and yet be precisely transmitted to the bellows.

DEVIATIONS IN TANK MEASUREMENTS

A vexing situation always arises when a fuel-oil purchaser is unable to reconcile the readings of his gauging with the oil distributor's statement of delivery. It is very seldom indeed that

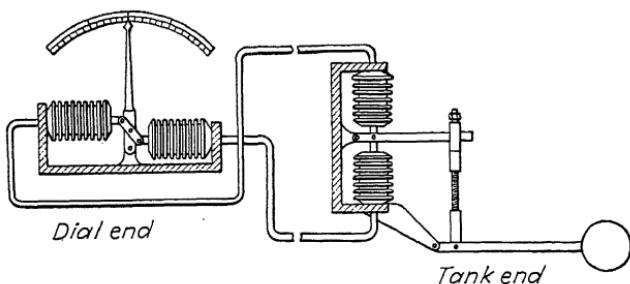


FIG. 200.—Combination float-and-hydrostatic gauge. (Liquidometer.)

there is any intent of fraud on either side, but as surely as both parties maintain the exactness of their measurements, it is equally certain that one of them is wrong. Experience shows that the benefit of the doubt should go to the distributor, for in by far the greater percentage of those cases where the volumes involved are actually verified, the distributor has been right. There are good reasons why this should be so, as a little reflection will show. The distributor's methods of measurement are closely supervised and rigidly checked by a municipal authority, and aside from all considerations of intentional shortaging which we may dismiss as of infrequent occurrence, it follows that a company whose very existence is predicated upon measurement and delivery of liquids will take especial care in assuring the preciseness of those measurements. As against this, the purchaser has only his own experience to guide him; there is no verification of the accuracy of his

gauging methods by any competent outside authority and there are a great many factors which may disturb his otherwise fairly accurate equipment.

In the case of rectangular tanks, the sides and ends usually buckle outward or "breathe," *i.e.*, the tank plates become convex by bending, thereby increasing the tank's actual capacity. In cylindrical horizontal tanks, the ends often behave in a similar fashion, although the effect is less marked. Discrepancies in depth measurement can easily lead to errors as high as 3 or 4 per cent by reason of tank breathing.

Underground tanks almost always sink after installation, usually one end settling more than the other, and buried tanks are generally set at an angle to the horizontal, in any event, so it is scarcely to be expected that the indicated depth of oil at any one spot in the tank will reflect a true measurement of the depth prevailing throughout the tank. Occasionally, too, tanks rotate longitudinally, with the result that neither a stick thrust down from the top nor an air chamber originally placed on the bottom for a hydrostatic gauge will show the actual depth of oil. As for any float device, it will probably be rendered totally inoperative by the stress placed upon it when moved from a true vertical position.

The actual total capacity of a fuel-oil tank is also subject to question. Because of the technical difficulties in fabrication, these tanks cannot be made to exact dimensions, and in fact the Underwriters permit deviations of as high as 10 per cent from the intended content. Hence a scale on an indirect gauge, or the markings on a gauge stick, will be inexact unless specially calibrated for the actual tank on which used.

A fairly close calculation of gallons per inch of depth of a rectangular tank can be made from the tank dimensions from the ratio of 231 cu. in. per U. S. gal. (277.274 cu. in. per imperial gal.; 5 imperial gal. equaling 6 U. S. gal.). Computations on cylindrical tanks are much more complicated, and the general practice is to resort to the use of tables of one sort or another whenever such calculations must be made. Such tables are given in the Appendix of this book. Should such an occasion arise, however, requiring a calculation of tank contents from the dimensions of the tank and the depth of liquid in it, the formula and coefficients in Table XLV afford the means:

TABLE XLV.—AREAS OF CIRCULAR SEGMENTS FOR RATIOS OF RISE AND DIAMETER*



$$\text{Area} = d^2 \times \text{Coefficient}$$

<i>b/d</i>	Coeffi- cient								
.001	.000042	.051	.015119	.101	.041477	.151	.074590	.201	.112625
.002	.000119	.052	.015561	.102	.042081	.152	.075307	.202	.113427
.003	.000219	.053	.016008	.103	.042687	.153	.076026	.203	.114231
.004	.000337	.054	.016458	.104	.043296	.154	.076747	.204	.115036
.005	.000471	.055	.016912	.105	.043908	.155	.077470	.205	.115842
.006	.000619	.056	.017369	.106	.044523	.156	.078194	.206	.116651
.007	.000779	.057	.017831	.107	.045140	.157	.078921	.207	.117460
.008	.000952	.058	.018297	.108	.045759	.158	.079650	.208	.118271
.009	.001135	.059	.018766	.109	.046381	.159	.080380	.209	.119084
.010	.001329	.060	.019239	.110	.047006	.160	.081112	.210	.119898
.011	.001533	.061	.019716	.111	.047633	.161	.081847	.211	.120713
.012	.001746	.062	.020197	.112	.048262	.162	.082582	.212	.121530
.013	.001969	.063	.020681	.113	.048894	.163	.083320	.213	.122348
.014	.002199	.064	.021168	.114	.049529	.164	.084060	.214	.123167
.015	.002438	.065	.021660	.115	.050165	.165	.084801	.215	.123988
.016	.002685	.066	.022155	.116	.050805	.166	.085545	.216	.124811
.017	.002940	.067	.022653	.117	.051446	.167	.086290	.217	.125634
.018	.003202	.068	.023155	.118	.052090	.168	.087037	.218	.126459
.019	.003472	.069	.023660	.119	.052737	.169	.087785	.219	.127286
.020	.003749	.070	.024168	.120	.053385	.170	.088536	.220	.128114
.021	.004032	.071	.024680	.121	.054037	.171	.089288	.221	.128943
.022	.004322	.072	.025196	.122	.054690	.172	.090042	.222	.129773
.023	.004619	.073	.025714	.123	.055346	.173	.090797	.223	.130605
.024	.004922	.074	.026236	.124	.056004	.174	.091555	.224	.131438
.025	.005231	.075	.026761	.125	.056664	.175	.092314	.225	.132273
.026	.005546	.076	.027290	.126	.057327	.176	.093074	.226	.133109
.027	.005867	.077	.027821	.127	.057991	.177	.093837	.227	.133946
.028	.006194	.078	.028356	.128	.058658	.178	.094601	.228	.134784
.029	.006527	.079	.028894	.129	.059328	.179	.095367	.229	.135624
.030	.006866	.080	.029435	.130	.059999	.180	.096135	.230	.136465
.031	.007209	.081	.029979	.131	.060673	.181	.096904	.231	.137307
.032	.007559	.082	.030526	.132	.061349	.182	.097675	.232	.138151
.033	.007913	.083	.031077	.133	.062027	.183	.098447	.233	.138996
.034	.008273	.084	.031630	.134	.062707	.184	.099221	.234	.139842
.035	.008638	.085	.032186	.135	.063389	.185	.099997	.235	.140689
.036	.009008	.086	.032746	.136	.064074	.186	.100774	.236	.141538
.037	.009383	.087	.033308	.137	.064761	.187	.101553	.237	.142388
.038	.009764	.088	.033873	.138	.065449	.188	.102334	.238	.143239
.039	.010148	.089	.034441	.139	.066140	.189	.103116	.239	.144091
.040	.010538	.090	.035012	.140	.066833	.190	.103900	.240	.144945
.041	.010932	.091	.035586	.141	.067528	.191	.104686	.241	.145800
.042	.011331	.092	.036182	.142	.068225	.192	.105472	.242	.146656
.043	.011734	.093	.036742	.143	.068924	.193	.106261	.243	.147513
.044	.012142	.094	.037324	.144	.069626	.194	.107051	.244	.148371
.045	.012555	.095	.037909	.145	.070329	.195	.107843	.245	.149231
.046	.012971	.096	.038497	.146	.071034	.196	.108636	.246	.150091
.047	.013393	.097	.039087	.147	.071741	.197	.109431	.247	.150953
.048	.013818	.098	.039681	.148	.072450	.198	.110227	.248	.151816
.049	.014248	.099	.040277	.149	.073162	.199	.111025	.249	.152681
.050	.014681	.100	.040875	.150	.073875	.200	.111824	.250	.153546

TABLE XLV.—AREAS OF CIRCULAR SEGMENTS FOR RATIOS OF RISE AND DIAMETER.—(Continued)

<i>b/d</i>	Coeffi- cient								
.251	.154413	.301	.199085	.351	.245935	.401	.294350	.451	.343778
.252	.155281	.302	.200003	.352	.246890	.402	.295330	.452	.344773
.253	.156149	.303	.200922	.353	.247845	.403	.296311	.453	.345768
.254	.157019	.304	.201841	.354	.248801	.404	.297292	.454	.346764
.255	.157891	.305	.202762	.355	.249758	.405	.298274	.455	.347760
.256	.158763	.306	.203683	.356	.250715	.406	.299256	.456	.348756
.257	.159636	.307	.204605	.357	.251673	.407	.300238	.457	.349752
.258	.160511	.308	.205528	.358	.252632	.408	.301221	.458	.350749
.259	.161386	.309	.206452	.359	.253561	.409	.302204	.459	.351745
.260	.162263	.310	.207376	.360	.254551	.410	.303187	.460	.352742
.261	.163141	.311	.208302	.361	.255511	.411	.304171	.461	.353739
.262	.164020	.312	.209228	.362	.256472	.412	.305156	.462	.354736
.263	.164900	.313	.210155	.363	.257433	.413	.306140	.463	.355733
.264	.165781	.314	.211083	.364	.258395	.414	.307125	.464	.356730
.265	.166663	.315	.212011	.365	.259358	.415	.308110	.465	.357728
.266	.167546	.316	.212941	.366	.260321	.416	.309096	.466	.358725
.267	.168431	.317	.213871	.367	.261285	.417	.310082	.467	.359723
.268	.169316	.318	.214802	.368	.262249	.418	.311068	.468	.360721
.269	.170202	.319	.215734	.369	.263214	.419	.312055	.469	.361719
.270	.171090	.320	.216666	.370	.264179	.420	.313042	.470	.362717
.271	.171978	.321	.217600	.371	.265145	.421	.314029	.471	.363715
.272	.172868	.322	.218534	.372	.266111	.422	.315017	.472	.364714
.273	.173758	.323	.219469	.373	.267078	.423	.316005	.473	.365712
.274	.174650	.324	.220404	.374	.268046	.424	.316993	.474	.366711
.275	.175542	.325	.221341	.375	.269014	.425	.317981	.475	.367710
.276	.176436	.326	.222278	.376	.269982	.426	.318970	.476	.368708
.277	.177330	.327	.223216	.377	.270951	.427	.319959	.477	.369707
.278	.178226	.328	.224154	.378	.271921	.428	.320949	.478	.370706
.279	.179122	.329	.225094	.379	.272891	.429	.321938	.479	.371705
.280	.180020	.330	.226034	.380	.273861	.430	.322928	.480	.372704
.281	.180918	.331	.226974	.381	.274832	.431	.323919	.481	.373704
.282	.181818	.332	.227916	.382	.275804	.432	.324909	.482	.374703
.283	.182718	.333	.228858	.383	.276776	.433	.325900	.483	.375702
.284	.183619	.334	.229801	.384	.277748	.434	.326891	.484	.376702
.285	.184522	.335	.230745	.385	.278721	.435	.327883	.485	.377701
.286	.185425	.336	.231689	.386	.279695	.436	.328874	.486	.378701
.287	.186329	.337	.232634	.387	.280669	.437	.329866	.487	.379701
.288	.187235	.338	.233580	.388	.281643	.438	.330858	.488	.380700
.289	.188141	.339	.234526	.389	.282618	.439	.331851	.489	.381700
.290	.189048	.340	.235473	.390	.283593	.440	.332843	.490	.382700
.291	.189956	.341	.236421	.391	.284569	.441	.333836	.491	.383700
.292	.190865	.342	.237369	.392	.285545	.442	.334829	.492	.384699
.293	.191774	.343	.238319	.393	.286521	.443	.335823	.493	.385699
.294	.192685	.344	.239268	.394	.287499	.444	.336816	.494	.386699
.295	.193597	.345	.240219	.395	.288476	.445	.337810	.495	.387699
.296	.194509	.346	.241170	.396	.289454	.446	.338804	.496	.388699
.297	.195423	.347	.242122	.397	.290432	.447	.339799	.497	.389699
.298	.196337	.348	.243074	.398	.291411	.448	.340793	.498	.390699
.299	.197252	.349	.244027	.399	.292390	.449	.341788	.499	.391699
.300	.198168	.350	.244980	.400	.293370	.450	.342783	.500	.392699

CONCRETE TANKS

By far the greater number of oil storage tanks installed for fuel-oil users are made of steel. It has been found that steel is the most practical material not only for small- and average-sized

storage, but also for large storage capacities. In some instances, however, concrete has been successfully used for bulk storage tanks. Concrete tanks have been found generally successful or practical for small and moderate capacities.

Among the regulations of the National Board of Fire Underwriters covering concrete tanks, will be found these stipulations:

1. Concrete may be used as material for construction for storage tanks when the fuel oil is heavier than 35 A.P.I.
2. Selection of sites shall be governed by rules for steel tanks.
3. All concrete tanks must be provided with a gastight concrete roof.
4. Rules for piping shall in general be the same as for steel tanks, except that it is preferred that piping pass through the roof if possible.
5. Plans for concrete tanks must be approved by the Inspection Department of the National Board before construction is started, and the entire erection must be under the charge of a fully qualified, competent engineer.

The Portland Cement Association suggests that the maximum working stress of concrete in storage tanks shall not exceed 8000 to 10,000 lb. The concrete shall always be reinforced with steel, as concrete does not possess the strength in tension which it has in compression, and steel must supplement the tensile strength. It is preferable that the steel be applied as hoops on the outside of the tank, the two ends being joined by conventional turnbuckles. This is better than placing the steel within the form and embedding it in the concrete, for the alternate stresses and strains of loading and unloading the walls have the effect of loosing the steel from the concrete, with resultant cracking of the concrete. Necessarily, this method of construction implies outside, aboveground, vertical cylindrical tanks. It is doubtful if other applications of concrete tanks will be permanent, effective or successful.

CHAPTER XVIII

SERVICE AND MAINTENANCE OF DOMESTIC AND COMMERCIAL OIL BURNERS

A good oil-burner-service mechanic must be somewhat like a country doctor—he must be prepared for handling any type of trouble that may arise. Specialization is as impossible with the one as with the other. The expert oil-burner-service mechanic must be skilled in, and able to take care of, problems arising in electricity and electrical apparatus, oil-burner construction and adjustment, combustion and flame application, refractories, heating plants and apparatus and a host of minor topics, such as sound insulation, plumbing and building construction. At the same time he will find it exceedingly helpful to himself, his employer and the customer if he has the instincts of a salesman and a diplomat.

ELECTRIC CIRCUITS

First in frequency of occurrence are trouble calls of an electrical nature, involving either a control, or the circuit, or the motor or other electrical apparatus. The mechanic should be equipped with the necessary apparatus for testing and if possible correcting these defects. Figure 201 shows various types of electric service distribution within a building. There are, in general, two operations for checking the condition of a circuit at any particular point. A jumper can be connected across the terminals of a control to determine whether the control is conducting current through. Or a test lamp or voltmeter or other high-voltage tester can be connected from the line to ground to determine whether the line is "hot" at the point being tested. In emergencies, should a control circuit be very complicated and it is desired to get power to the motor, oil valve and ignition quickly without regard to automatic operation, then the simplest procedure is to run temporary wires direct from the service switch down to the boiler front, disconnecting any wires to the per-

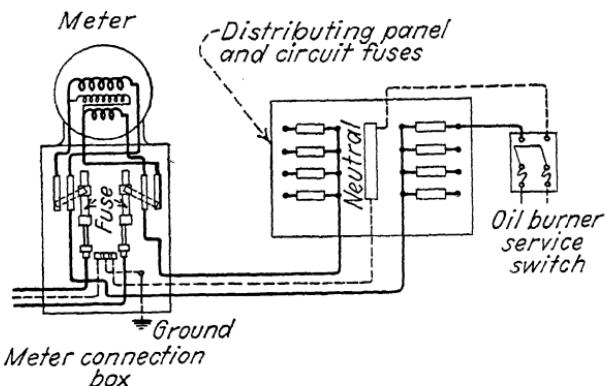


FIG. 201a.—Meter connections and distribution panel for 110-volt alternating current, single phase, or 110-volt direct current, grounded neutral.

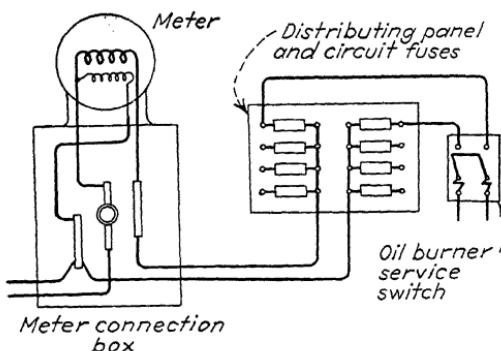


FIG. 201b.—Meter connections and distribution panel for 220-volt alternating current, single phase, or 220-volt direct current.

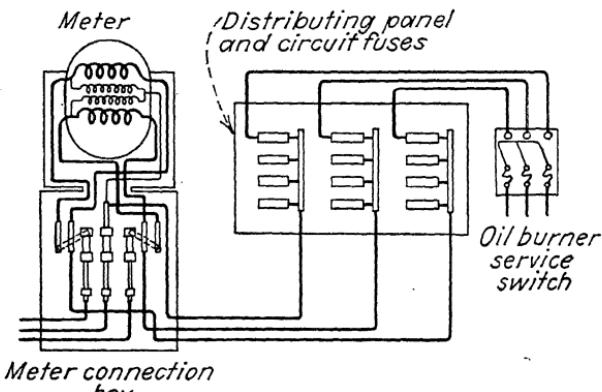


FIG. 201c.—Meter connections and distribution panel for 220-volt alternating current, three-phase.

manent circuit which would interfere with this type of temporary operation.

Most domestic installations operate on 110-volt single-phase power. Here one line is usually hot, or at a potential of 110 volts from the ground or environment, while the other line is a ground line, or at the same potential as the environment. It is a general practice to use black wire for the hot line and white wire for the ground. If the power is 220 volt single phase, both lines are hot and at a potential of 110 volts to ground, but a potential difference of 220 volts exists between the two wires. In testing such lines, an ordinary light bulb in a socket may be used to connect either line to ground, but to test across the two lines requires two such bulbs in a series, or the equivalent. In the case of three-phase power, somewhat varying services may be encountered. Ordinarily, however, it will be found that all three lines carry a voltage of 110 volts to ground, but 220 volts between any two of the three wires (or close to 220 volts). Two-phase systems may be either three wire or four wire. In the latter, the service is really the same as two independent single-phase lines, there merely being two sets of two wires each acting just as above described for 220 volt single phase. In the three-wire type, one wire serves as a common to the other two, there thus being one phase from the common to each of the other two, but there being no potential across the two independent lines. There are also other types of two phase, involving more than four wires, but such are not very common. Single and three phase are by far the predominating systems encountered in American practice. Direct current is also found, though not nearly so frequently or extensively as alternating. Direct current 110 volts is almost identical with 110 volt single phase, as is 220-volt direct current with 220 volt single phase, insofar as potentials to ground and between wires are concerned. On both alternating and direct current, voltages exceeding 220 volts are sometimes used, 440 not infrequently. But voltages higher than 440 are rarely encountered in oil-burner applications, as is also true of odd voltages running in other than multiples of 110.

A common service necessitated on automatic controls is on the electrical contacts. There are two types of contact, the so-called open contact and the mercury tube. Either type can be applied to either low voltage or line voltage provided the design is correct

for the current carried. Usually, open contacts intended for line voltage are moved by mechanisms which bring the contacts into closed position by snap action, and again break them apart by a similar sharp movement. This avoids bridging of the current across a narrow gap, which arcing rapidly corrodes the surfaces of contact. Whenever the contact surfaces have become pitted, whether by normal wear over a long period or by rapid corroding due to an arc from defective making and breaking, the surfaces must be made smooth and even by very gentle and careful filing, as with a nail file or fine sandpaper or emery cloth. Low-voltage open contacts also become pitted, occasionally, and must be treated the same way, but with much more gentleness, using a very fine abrasive such as talc powder. Mercury-tube contacts will usually remain in working condition so long as the tube remains intact, but should a crack occur in the glass and so permit the entrance of air, then the mercury oxidizes rapidly and clings to the contact prongs instead of leaving with a clean break when the tube assumes the off position. A new tube must then be inserted to replace the bad one. Defective switching by an electrical control can usually be traced down to the place at fault when the burner operates improperly.

The control mechanisms which operate the contacts are in turn actuated by variations in temperature or pressure (of air, steam, water, etc.) or by other electrical devices (as solenoid coils, etc.). Usually all such instruments are accompanied by written instructions from the manufacturer for adjusting the operating differentials or points of make and break. If such cannot be had when it becomes necessary to work on an instrument, then it will generally be found that there are two sets of adjustments: one for varying the differential or interval between opening and closing of the switch, and one for setting either the cut-in or cutout temperature or pressure. The setting of cut-in point and differential therefore determines the cutout point; conversely, cutout and differential settings determine the cut-in point. Manufacturers make these adjusting devices as simple as possible, either springs, bellows, diaphragms or similar contrivance, which can be adjusted by turning of a screw or nut so as to vary the tension or elasticity of the device. Handling of control instruments should always be done with care and precision to avoid injury to the delicate mechanisms and to assure that the desired adjust-

ments are performed correctly. An instrument should always be checked by allowing it to perform normally through several cycles of operation, after an adjustment of its setting.

The two most commonly used motors in domestic oil burners are the repulsion-induction and the split phase; in commercial burners the same two are used extensively, with the squirrel cage the favorite if three-phase power is available. In direct-current districts, the shunt and compound motors are most generally used. None of these motors is extremely complex, although undoubtedly the squirrel cage is the most simple since it has no windings on the rotor, no brushes and but one stator winding. The other two alternating-current motors both have starting windings as well as running windings; in the repulsion-induction it is located in the rotor, in the split phase, it is in the stator. In the case of both repulsion-induction and split phase, should the starting winding burn out, but the running winding remain intact, then, although, of course, the motor will not start under its own power, it is often possible to bring it up to speed (as by wrapping a string around the shaft at some point outside the frame and then pulling the string in much the same manner that a top would be spun) and then, by throwing in the current to the running winding, to keep the motor running until repairs or replacement can be arranged for. Brush replacements are the most frequent service requirement on repulsion-induction or direct-current motors, although if commutators and windings are well kept a set of brushes should give many months of continuous service. To reverse rotation of electric motors on split phase, change the line connections to either the running or starting windings, but not to both; on repulsion-induction, shift the position of the brush holder slightly until a point is found at which the motor starts up in the desired direction; on squirrel cage, reverse any two of the three line connections; on direct current, reverse the line connections to either the field or armature, but not to both. If the motor does not come up to full speed and the shaft is free on split-phase and squirrel-cage motors the difficulty is most likely to be found in the rotor, and very probably the solder connections between segments are open; on repulsion-induction it may be an open in the rotor winding or poor contact of the brushes against the armature, either because the brushes are worn or because the commutator is dirty; with direct-current motors the

case is more complicated, and it is best to call a motor expert promptly or to install another motor and send the bad one to the shop. Of course, in all cases where a motor, regardless of type, will not come to speed when first placed into service, the voltage, phase and frequency should be checked. It is equally obvious that a shaft which binds and does not turn freely will slow down the motor.

SERVICE ON OIL PUMPS

The rotary gear-type pump is used almost exclusively in domestic and commercial burners. Made in a variety of styles and forms, there is still great uniformity in design, the internal gear type being by far the most popular, although the external gear is by no means uncommon. Gear pumps are exceedingly dependable in operation, requiring but little maintenance and service. Possibly 75 per cent of all service calls involving the pump will be found to be air-binding, and this is the fault of a leaky suction line rather than the pump itself. Whenever it is found that the pump is not circulating oil, it is good practice to check almost all other features of an installation which could be contributing factors rather than to suspect the pump itself. However, if it be established that the suction line is tight and the strainers are clear and that all valves are open and that there is fuel in the tank, then the pump can be examined. Naturally, the first point is to see that the motor is really turning the pump and that the coupling or belt or other driving contrivances are not at fault. Next the direction of rotation should be verified. If the pump shaft passes through a packing gland, this must be airtight; sometimes the packing nut must be turned in to tighten the packing. If finally, all these points have been checked and found all right, then, especially if the pump is fairly old, the pump can be taken apart for examination of the parts. The head gasket is sometimes found to be defective. Of course, it is also possible that the gears may have worn and are no longer making close enough mesh to establish a suction. A quick method of determining the probable condition of the pump is to place a vacuum gauge in the line as close as possible to the inlet port, and then close the suction line with a valve or by capping up the line itself. If then it is known that the pump is primed, the pump can be started. A good pump under these conditions should pull

a vacuum close to within 1 in. of the perfect vacuum, *i.e.*, at sea level about 28 or 29 in., and at higher elevations, the proportional value. When the pump fails to do this, and it is definitely known that no other factor can be contributing, then the pump should be replaced.

If the pump shows a good vacuum on the above test and is still unable to pull oil over, then, of course, the trouble lies in the suction line (assuming, of course, that the tank contains oil to the height of the suction stub). Should the vacuum gauge show a high reading while the pump fails to draw oil, then the suction line is plugged (and the chances are the trouble is in the strainer); should the vacuum be low or zero, then the line is open to atmosphere at some point. The most likely places to look for air leaks are the caps of strainers, pipe unions, stuffing glands on valves and pipe joints. Very often an application of paint or shellac to suspected points on the line will close up the leak, if it be a small one. When trying to pull oil up into an empty suction line, it is helpful to vent the pump discharge line at a point close by, to assure prompt and adequate release of air which might otherwise hamper the pump in its attempts to establish suction. In trying to verify the existence of a suspected air leak, when the pump is circulating oil, if a plug or pressure gauge in the discharge pipe be removed to permit the oil to come up through, the presence of air in the oil will be indicated by air bubbles or froth in the oil.

OIL FEED

The oil-flow system on the pressure side of the pump also gives rise to occasional failures in service and operation. Circulation may be established to and through the pump, and the oil-pressure gauge may show ample oil at the burner; still there may be no delivery of fuel to and through the nozzle. Usually, difficulties on this side are more readily located and corrected. Most commonly of all, the nozzles become plugged. Then, the line to the nozzle may be stopped up, or a strainer fouled. A valve might inadvertently have been closed off. Where a magnetic oil valve is employed, the plunger may not be rising, owing either to failure of the coil or interruption of electric circuit to the coil or because the oil pressure exceeds the maximum against which the coil can cause the plunger to rise. The converse to no oil can also occur on the pressure side of the pump, *i.e.*, the oil may continue to

run after the burner has shut down. This could be caused by dirt under the seat of the magnetic oil valve, preventing the plunger from seating tightly. Or the pressure-regulating valve of a high-pressure atomizing burner could be defective, failing to shut off the oil flow to nozzle when the pressure drops to the point at which it should normally shut off. The latter condition may be corrected by adjusting the mechanism, or the valve or valve part may need repair or replacement.

Oil strainers do not give rise to a great deal of service, although, of course, they must be kept clean. How often a strainer should be cleaned depends entirely on the sediment content of the oil being strained and on the amount of filtering surface in the strainer. After a strainer has been opened for cleaning it is extremely important that the cover be replaced in such manner that it makes an airtight seal with the strainer body. In some types, where a gasket is used, the gasket must be replaced with a new one each time the strainer is opened. Although not a common occurrence, there have been occasions when a strainer body has cracked, resulting in air leaks into the suction line. Should this occur, of course, it must be located, and as a rule it is about the last thing that a mechanic looks for when seeking an air leak. If a strainer is suspected of leaking air, the surest and quickest way to verify it, if the crack is not visible, is to place the strainer under pressure. Tiny holes in the casting can also be located in this manner.

OIL-PRESSURE REGULATORS

There are two types of pressure-regulating apparatus, which might loosely be classified as low pressure and high pressure, although this is not strictly a distinguishing feature since the high limit of the low-pressure type approaches and sometimes reaches the low limit of high-pressure type. In general, the low-pressure regulator (commonly used on commercial burners such as the horizontal rotary), is termed a pressure-relief valve, and usually functions to maintain oil pressures between 5 and 25 lb. per sq. in., although values of 50 lb. are sometimes needed in the burners using these regulators, and, though rarely, an unusual operating condition may send the oil pressure as high as 75 to 100 lb. The second group, the so-called pressure-regulating valves, are characteristic of the gun-type high-pressure

atomizing burners, and are called upon to maintain pressures averaging around 100 lb., with 60 as the minimum and 200 as the maximum of the operating ranges.

The pressure-relief valve almost universally uses a spring as the pressure-generating mechanism, which spring creates tension against a plunger. The pump forces oil up against the plunger, and in this manner the tension of the spring determines how much pressure the pump must develop to force oil through the valve. In operation, there can be two kinds of trouble arising: no-pressure and low pressure. Whenever it is found that the valve is creating little or no pressure, the most likely cause will be dirt or scale which has become lodged on the seat of the valve and which prevents the spring from forcing the plunger down close to the seat. Or the plunger may have become stuck, thereby rendering the valve inoperative. In either case the remedy is simple and obvious. Low pressure can also be the result of either of two defective conditions: the valve seat may be scored, permitting oil to leak through in greater amount than the clearance between the plunger and seat would allow; and the spring may have lost its tension or elasticity. To correct the first condition, the seat must be ground smooth, in much the same manner as is done with automobile engine valves. The simplest procedure in the second case is to replace the spring, although temporary repairs may be made, if a new spring is not immediately available, by stretching the old one and recompressing it into the valve chamber.

With the high-pressure regulator, no-pressure will result from the same source as above, dirt or scale on the seat, or the plunger binding. Low pressure is a rare occurrence in this type of valve, although sometimes in that particular form of valve which employs a linkage mechanism to transmit tension to the valve seat, the linkage may be working irregularly, with the result of either no-pressure or low pressure. Generally, it will be found easiest to replace the valve entirely, taking the old one to the shop for inspection and test, where it may be either repaired or discarded. Should oil be found to be leaking out of the valve, it will in all likelihood be caused by a cracked bellows or diaphragm. These valves usually also incorporate a feature for shutting off oil flow to the nozzle when the pressure falls below the safe minimum for good atomization. Should oil continue to dribble

through the nozzle after the pump stops running, it will almost invariably be caused by air in the oil, which air causes the shutoff feature to become inoperative. The air, of course, is brought in through a leak in the suction line, which must be located and stopped. A scored seat in the shutoff part of the regulator will also cause drooling; a condition which can sometimes be corrected by inserting a new seat if it be of the replaceable type, but usually a complete valve replacement is required.

ATOMIZERS AND NOZZLES

Atomizers and nozzles, if they receive intelligent handling at such times as they require routine attention for cleaning, should give very little operating difficulty. Properly cared for, these parts will give years of satisfactory service. When they do require unusual attention or replacement after but a relatively short period of use, it is either because some very unusual condition obtains in a particular installation or because a slipshod method has been used in handling the parts. So far as atomizers of horizontal rotaries are concerned, the chief forms of abuse are: nicking or bending the atomizing edge, as by dropping or pounding, and putting the atomizer out of round, which also occurs from striking it. Should carbon formations be deposited around the atomizer while in use, frequently the high heat reflecting back onto the atomizers softens the metal with resultant change in its form. Atomizers should always be cleaned by the aid of soft cloths and solvents such as kerosene, rather than harsh scraping with a metallic instrument. It is true that atomizers sometimes gradually shorten up, so that after a long period of use the edge has receded to a point inside the air-nozzle periphery. At such time the atomizer is no longer fit for use and a new one should be installed. In the case of vertical-rotary domestic burners, the same remarks are applicable. In the vertical rotary, true atomizer shape is even more important, for the high speeds of rotation which some of these burners employ will make, of course, an unbalanced cup cause excessive bearing wear, noise and carbon. With pressure-atomizing nozzles, anyone who is at all familiar with the extremely fine machine work represented by the orifices and slots will appreciate why delicate care must be used at any time that the nozzle is removed from the burner for cleaning. Absolute cleanliness is imperative, and it simply is beyond

consideration that any sort of hard or sharp instrument be applied to such nozzle to clean it. Abusive handling results in deformed orifices and slots, which will be manifested in operation by poor atomization, smoke and carbon formation. When a nozzle has been so badly used as to have imperfect orifice or deformed slots, there is but one remedy—a new nozzle tip.

DIFFICULTIES ARISING FROM POOR OR DEFECTIVE INSTALLATION

The most obvious ill effect of poor or faulty workmanship in making installation of an oil burner will be in the fire itself, and will be indicated by a variety of symptoms, such as smoke, carbon formation, inability to carry a clean fire or a large enough fire, or failure of the firebox to heat sufficiently or uniformly, as well as by combustion roar or noise and sooted flues. With any of these or related defects in operation, the installation should be systematically checked against the following summary of points.

Defective Condition	Operating Difficulty Resulting
Firebox too short.	Carbon on backwall, black smoke.
Firebox too long.	Cold refractory, soot in flues, white or blue smoke from stack.
Firebox too narrow.	Carbon on side walls.
Firebox too wide.	Cold side walls, ragged and smoky fire.
Firebox tapers too sharply at front.	Carbon on refractory near atomizer.
Backwall too high, flame baffled too much.	Pulsing or rolling fire, smoke.
Setting not airtight.	Low carbon dioxide, wasteful combustion.
Nozzle too close to floor.	Carbon on floor, irregular fire due to poor mixture with air, dirty fire.
Nozzle too high above floor.	Cold floor, smoldering oil deposits.
Nozzle too near crown sheet of boiler.	Very sooty flues.
Secondary air port too small.	Black smoke, choking fire, gasping.
Secondary air port too large.	Blue or white smoke from stack.
Secondary air port not located properly.	Poor mixture air and oil, smoke and inability to carry load.
Primary air too great, shutter open too wide.	Noisy, roaring fire.
Primary air inadequate, shutter not open enough.	Smoky fire, smoke from stack.
Burner improperly sized, not enough capacity.	Smoke, inability to carry load.
Oil piping too small.	Inability to secure large enough fire.
Oil suction pipe leaking air.	Fire pulsates and flutters.

Sources of noise other than combustion roar (treated above).

Oil pump out of line.

Some rotating part, as fan or blower wheel, rubbing on housing.

Some rotating part, as fan or motor armature, out of balance.

Worn bearings.

Loose or improperly hung piping.

Air passing through relief or regulating valves.

High vacuum caused by dirty strainers, or other restrictions

Motor hum, or magnetic or solenoid coil hum, or poor electrical or mechanical contacts in relay, chattering.

DIFFICULTIES ARISING DURING ROUTINE OPERATION

First honors for frequency of occurrence probably go to the sooted stackswitch, or combustion safety switch. Whichever type be used, the one with the bimetal coil or the disk, both are dependent for their proper function upon the heat-sensitive qualities of the element which in the first type is located in the path of the hot gases and in the second is located so as to sight upon the fire. When a coating of soot has been laid down, the instrument loses its sensitivity to heat. The result, of course, is interrupted operation of the burner, for the combustion switch then acts as though there were a flame failure, and the burner "goes off on safety." Someone then has to push the reset button, after cleaning the stackswitch, and only too often it is the service mechanic who has been summoned instead of the owner or operator who should be conversant with this feature of his burner and be able to correct it when it occurs.

Another undesirable effect of soot or carbon concerns its deposition on the electrodes of either an electric or electric-gas ignition system. When a carbon deposit completely bridges across the gap and short-circuits the electrodes, then the only result is failure of ignition, with the burner stopping on safety. But when the carbon or soot formation merely delays the spark without completely eliminating it, then disaster may follow. For should the delay of a few seconds in ignition mean that the firebox will meanwhile fill up with oil fumes from the nozzle, with the spark coming on in so unfortunate a circumstance, then, of course, an explosion will occur—the violence and damage being determined by the nature of the oil, size of the combustion chamber, restriction to normal expansion of the gases up the stack, etc. Fortunately, it is seldom that this phenomenon causes more

than a dislodged chimney pipe from the boiler to the stack, or possibly a door blown open on the boiler front.

In the case of horizontal-rotary burners, especially when the fuel runs a bit too heavy, a carbon deposit forms on the air nozzle which surrounds the oil atomizer. This deposit will cut down the primary air supply if it forms radially inward, and as it approaches the cup the fire becomes smoky. Routine cleaning should always include the removal of any carbon or gum on the air nozzle, as well as any dirt or deposit in the cup itself.

When the service mechanic is shooting a call on high oil consumption, he will do well to check over: carbon dioxide or excess air, improper thermostat setting or operation resulting in overheating or heating during the night, poor heating plant performance, or undue heat loss from the building, as by lack of window stripping, etc.

By this time most oil-burner dealers, and even the general public, have been educated to the importance of draft control. Safety precautions dictate that the damper in the breeching be clamped or secured firmly in a wide open position, to avoid possibility of its closing entirely. While such a measure is wise, and is in fact required by the Fire Underwriters, it also gives the draft unhindered play, and may mean that large quantities of excess air are drawn through the firebox. There are available many designs of automatic draft controllers, and almost every oil-burner installation should be equipped with one. As a rule, the simpler the device is, the less likely it is to give trouble during operation. It should be remembered that once a draft controller is installed there will be little if any attention paid to it thereafter. Hence complicated mechanisms that require skilled supervision, such as might be suited to an industrial plant which is under the constant charge of a trained attendant, are generally not suited to small commercial, or domestic, oil burners.

ROUTINE OF MAINTENANCE

First and foremost, lubrication is the most important of maintenance functions. For while other features of an installation, such as dirty strainers or nozzles, will give evidence of their condition by disturbing the normal operation of the burner, bearings as a rule give no warning of their need for lubrication, nor do

gears or other working parts which require oil or grease. Generally, the first outward manifestation of neglected oiling or greasing is the last, insofar as any particular case is concerned, for a renewal of a bearing or gear is by that time probably the only remedy. Motor manufacturers have perfected bearing wells which give remarkable service, from the viewpoint that oiling is needed at only long intervals, and in most domestic burners, it is sufficient if the motor bearings be oiled only once each season, say at the beginning of the season when the burner is first placed in operation after the summer shutdown. But at that time the oiling operation should be performed. Pump gears in many commercial burners require closer attention, and usually the gear casings are provided with a means of verifying the adequacy of the oil in the reservoir. Recent developments have been towards packing the pump-drive gears in grease, so as to provide a supply that will be sufficient for a longer period of time than lubricating oil. But in all cases, the lubrication of pump-drive gears is of sufficient importance to warrant fairly frequent checking. Air compressors also need a goodly amount of oil, and well-designed compressors are equipped with an oil reservoir and lubricator which feeds it to the bearings uniformly and continuously, or else grease cups are used, of a type which can be refilled easily by someone not necessarily trained in mechanical maintenance.

Second in importance for routine servicing of burners is cleaning. Strainers, nozzles, and atomizers, fans, motors—in fact all parts which are subject to exposure, dirt and sediment should be carefully and thoroughly cleaned. Cleanliness is more than half the secret of successful burner performance.

Regular checking of the safety devices is also sound practice. Low-water cutouts should be put through a cycle of operation by draining water from the boiler till the switch trips. This should be done with the burner in operation, so that the test proves out, not only the cutout itself, but the complete electrical circuit and all the relays and contactors which are part of the circuit. The same should be done with pressurestats and other limit controls. It is a sensible thing also to try out the flame-failure safeties on the same call.

A final word on care of refractories should here be added. Refractories begin to deteriorate, by crumbling and disintegra-

tion, at those points where the surface becomes cracked, or where openings occur in the bonds between the bricks, for oil particles enter such crevices and subsequently char. The coke or carbon which is so formed expands and so breaks the refractory apart. All such breaks which happen in the original surface should be filled in with a high temperature cement, and the entire firebox surface should be kept coated with this cement so as to fill up the pores of the brick. Incidentally, all that has been said above in this chapter about avoidance of carbon formation is also helpful toward longer refractory life. When carbon forms it must be removed; the physical operation of breaking up the carbon formation with a poker or other contrivance is always harmful to the brick, and gradually, if it is continuous, will crack up the firebox.

In closing this chapter, it is perhaps most fitting to remind the service mechanic that his is the task of keeping the oil-burner user happy with his burner; that the most frequent contact of the user with the oil-burner industry is through the service mechanic; and that therefore it devolves upon him to keep the user posted on just what he should do for himself to secure the greatest benefit and usefulness from the oil burner.

Below is a brief outline of a complete analysis of a no-heat service call, based upon an installation with a magnetic burner relay of some type.

ANALYSIS OF TYPICAL NO-HEAT SERVICE CALL ON AN AUTOMATIC OIL BURNER

Condition on arrival of mechanic at building: Thermostat is calling for heat but burner is not running and building is cold (*i.e.*, burner could not be off on limit control).

Mechanic presses reset button. One of three things must happen:

- I. Burner will not start.
- II. Burner starts but no fire results.
- III. Burner starts, fire results, but burner stops again in few moments.

The following discussion shows how the difficulty can be systematically located by routine procedure.

- I. Burner will not start.
 1. Check fuses; if blown, replace. If fuses are all right, and line to relay is hot, then
 2. Remove cover of relay and push in or raise contacts to make circuit to motor. If motor starts, proceed as in *a*. If motor does not start, proceed as in *b*.

- a. For series 10 circuits, if relay stays in when pushed in but will not pull itself in, then look for open in blue wire circuit. If relay will not hold itself in, but motor runs when relay is manually held in, look for open in white wire circuit (if series 10), or in thermostat contacts or in line through controls (if line voltage). If relay stays in when pushed in, but ignition does not come on, combustion switch may be in hot position.
- b. Low-water cutout or other switch in line may be open, or wire may be broken in circuit. If switches are closed and wires are all right, check motor. Motor may be burned out, brushes may not be making contact, some wire in motor or brush leads may be open. Or motor may be stalled by frozen shaft, especially if it hums but will not run. If magnetic motor starter is used connection may be broken between relay and starter or pull coil of starter may be burned out or open. If overload relay is used in motor circuit, it may be open.

II. Burner starts, but no fire results.

1. There may be oil sprayed into the firebox, but no ignition occurs. If so, proceed as in a.
2. There may be no oil sprayed into firebox; proceed as in b.
 - a. If electric ignition, check the electrodes for carbon across the gap. If expanding-gas ignition, pilot may be out with gas flowing or gas may be shut off or pipe may be clogged. If electric-gas ignition, high tension cables may be grounded or open; or transformer may be burned out or otherwise not functioning. If ignition is on but oil will not ignite, it may be too heavy an oil, have too high a flash point, or contain much water. If electric or electric-gas ignition, the spark may be out of position. If expanding-gas ignition, plunger of magnetic gas valve may not be rising to permit full flow of gas. Ignition circuit from relay to transformer or gas valve or both may be open, or ignition contacts in relay may not be making.
 - b. Verify oil supply in tank, and then whether all valves in oil supply line are open. Verify direction of rotation of motor, and of pump, and whether pump is actually turning, and at right speed. Then check strainers which may be clogged up. Then check suction line for air leaks, especially around unions and strainer caps. Pump packing may be leaking air or pump may be air bound and require priming. Pump may be worn. If pump is circulating oil, but none is reaching nozzle, nozzle may be clogged up, or strainer between pump and nozzle may be clogged, or a valve may be shut off. Pump pressure may not be high enough.

III. Burner starts, fire results, but burner shuts down in a few moments.

1. The fire may be too small to actuate combustion switch. Increase size of fire.
2. The combustion switch may be coated with soot or otherwise shielded from heat of the fire. Clean, or otherwise correct.
3. The wires to combustion switch may be open, or the combustion switch connections may not be making. Rectify.
4. The heat sensitive element of the combustion switch may not be functioning. Repair or replace.

CHAPTER XIX

SHELL-TYPE VAPORIZING BURNERS

A comparatively recent development in the field of small oil burners is an adaptation of the vaporizing burner, heretofore variously known as the blue-flame burner, range or stove burner, and distillate burner, but now coming to be called the shell-type burner. None of these terms, except perhaps the last, is adequately descriptive or sufficiently distinctive to distinguish these burners entirely separately from all others, for the reasons that an important group of motor-driven domestic burners employ the vaporizing principle and are characterized by a blue flame within the firebox, and also operate on No. 1 or distillate fuel oil. Perhaps the single distinguishing feature of the burners referred to in this chapter is the fact that universally they operate on natural draft and employ no mechanical power of any kind for accomplishing atomization, whereas practically all other types of burner utilize either forced or induced draft or effect oil atomization through the agency of a motor-driven atomizer or pump or other mechanical device.

Blue-flame burners are built only in relatively small sizes and so are intended only for very low combustion rates. This permits of carrying on the vaporization within a restricted area, and of then confining the vapors within a very limited volume so that admixture with air and subsequent ignition can be attained with ease and effectiveness. The ensuing flame burns partly within the burner itself and partly in the free space above the burner, but is enclosed by the stove or heater.

Oil is fed by gravity into a shallow trough which is formed in the base plate of the burner proper or supported immediately above the base plate. The maintained oil level in the oil trough or pan is regulated by the elevation of the fuel supply, as described below. The oil pan is circular and contains also a wick which extends around the circumference of the trough and rises from the bottom to a point about 1 in. above the oil level. Thus the

wick becomes saturated with oil and presents an evaporating surface from which oil vapor may arise. The trough is also completely encircled by two perforated alloy-steel cylinders, the outer of which also encloses the wick and the inner of which completes the space into which the oil vapor must enter and rise through. Air is inspirated into the oil fumes by being drawn through the perforations of the steel cylinders, or, as they are commonly termed, shells. Combustion is initiated by applying

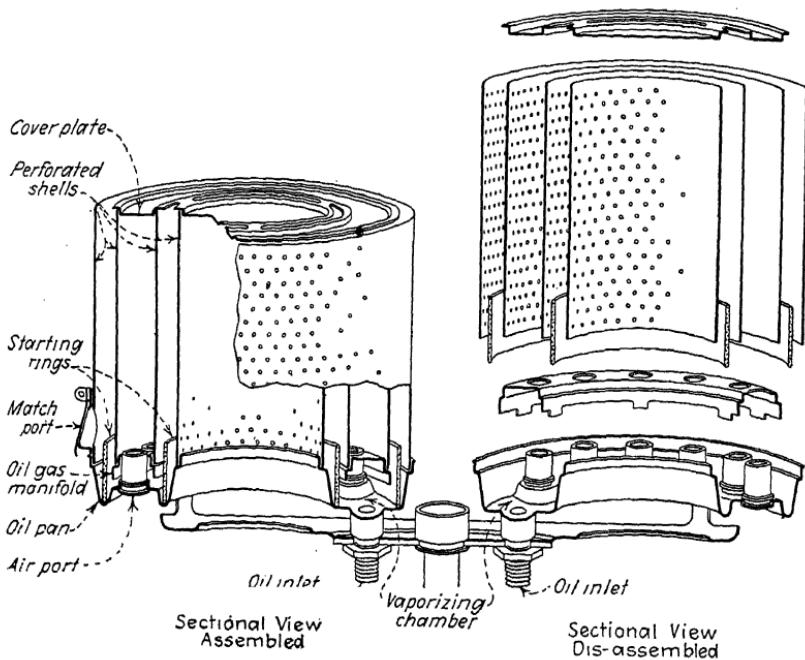


FIG. 202.—Construction of shell-type burner. (Courtesy of Automatic Burner Corporation.)

a match or torch to the wick. The flame begins burning at the wick, but as the combustion space becomes heated, the vapors are generated more rapidly and rise somewhat above the bottom. The flame may be said to start just above the wicks, and to assume increasing body as it progresses toward the top. Except for the lower $\frac{1}{2}$ in., the shells glow with a bright red.

Usually, there are at least two complete sets of concentric oil pans and steel shells, giving the effect of two combustion chambers, in a standard vaporizing burner of this type. Sometimes there are three and even four sets in the larger capacity burners.

And finally, two such burners, each consisting of two or more sets of shells, are usually grouped together to give a twin-burner unit, the oil flow to each being separately controlled through a twin-valve assembly. The twin-burner assembly can be set into the firebox of the average stove or range, its setting being adjustable as to height. Oil is supplied from bottles which are supported near by on an adjustable height stand. The valves can

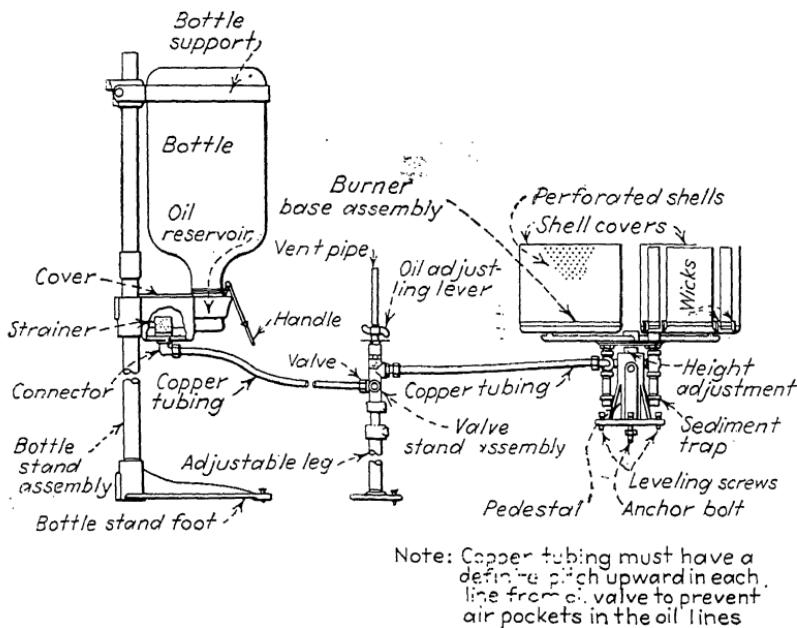


FIG. 203.—Diagram of oil-burning system for shell-type burner.

be supported either from the floor on their own stand or else be bolted to the stove itself.

A typical commercial form of such a burner is shown in Fig. 202. A pressed-steel base supporting the two burners is in turn held up in position on an adjustable pedestal. Each burner has two sets of two shells each, which fit into the grooves of the base. There are two oil troughs which connect into each other and into a gas manifold. Pressed-steel covers on top of the perforated shells are used both for spacing the shells and for directing the flow of air into the combustion spaces between the shells. Air openings between the troughs in the base provide a supply of preheated air between the shells.

The fuel-supply system comprises one or two bottles, each of 2 gal. capacity, inverted and supported with the necks immersed in an oil reservoir. Whenever the oil level in the reservoir falls

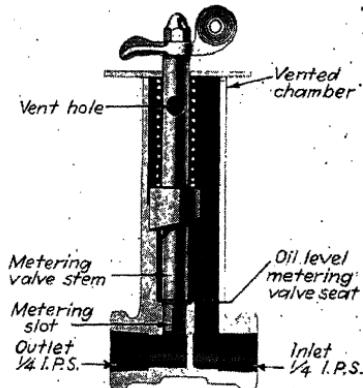


FIG. 204.—Special metering valve for gravity feed. (Courtesy of Automatic Products Company.)

below the level of the bottles' openings, air is admitted into the bottles and oil flows out. But immediately after sufficient oil leaves the bottles to raise the oil level in the reservoir above the bottle necks, the oil is again sealed within the bottles. From the oil reservoir, a copper tubing conducts oil to the oil-regulating valves, which are located near the stove. The tubing should slope downward between the reservoir and the valves, and again upwards between the

valves and burners, to allow air to rise upward in either direction and thus avoid difficulty with air pockets which otherwise tend to form in the tubing.

The oil-regulating valves are a very important part of the system. Uniform oil feeding, with easy regulation or change of rate of flow, must be made possible if satisfactory operation is to result from this type of apparatus. Ordinary needle valves are not sufficiently dependable for this work. Many different styles of valves have been evolved for this application, in which the opening through which the oil must flow is concentrated so that small dimensions are avoided. Special sleeves are provided in which a valve stem can be rotated so as to control the oil-flow opening laterally, the opening being in the shape of a notch or slot whose exposed aperture is thus made variable. In this manner (Fig. 204) clogging by deposition of sediment and formation of film across the valve opening is minimized.

When being installed, the twin unit is set up in the stove firebox, the grates having first been removed. The pedestal rests on the ashpit bottom, to which it is bolted for rigidity. The tops of the burners must be between 4 and 7 in. below the top of the

stove, to give ample combustion space. Should it be necessary, the burner tops may come to within 2 or 3 in. of the stove top, but this restricts the capacity of the burners by reducing the combustion volume. In addition, this closeness of the flame to the metal of the stove may increase the tendency toward carbon or soot formation. The burners must be set quite level to assure uniform

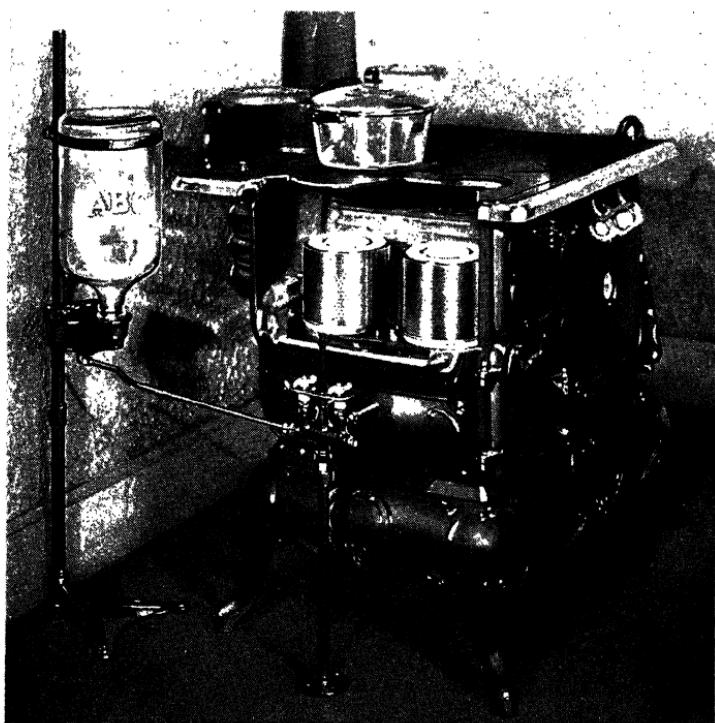


FIG. 205.—Typical installation of shell-type burners in kitchen range. (*Courtesy of Automatic Burner Corporation.*)

spreading of the oil throughout the troughs. The height of the oil reservoir above the floor should be very closely the same as the vertical distance between the oil troughs and the floor; in other words, the head of oil creating flow is very small, since the rates of oil flow are very low.

These burners require but very little draft. The damper in the chimney should have a hole, equal to 20 per cent of the chimney area, in its center, and then may be set in a position yielding

the best flame results. Usually, this damper may be almost completely closed except for the hole in the center. Very often it will be found necessary to install a balanced draft adjustor on the chimney to avoid excess draft. The heat created by the flame creates adequate draft through the burners to induce air flow in sufficient amount to supply air for combustion.

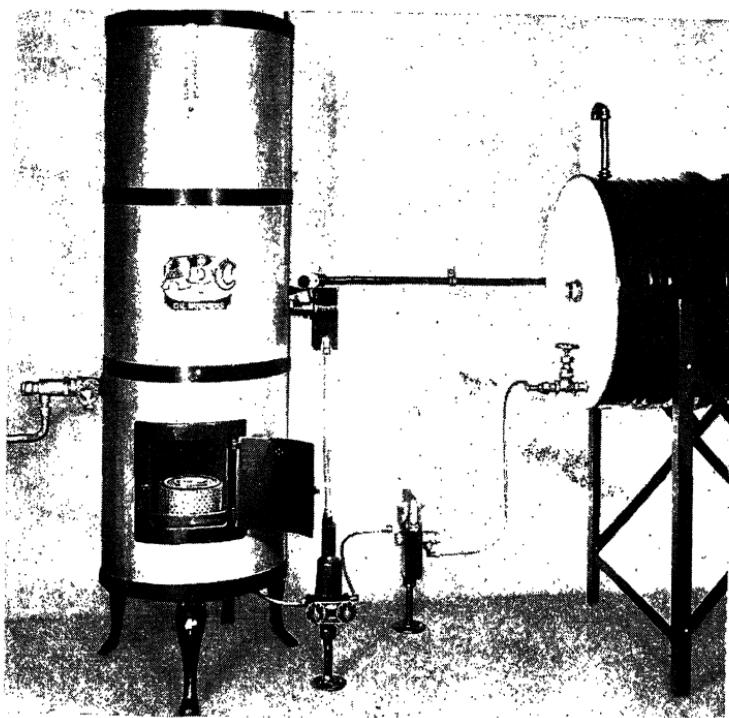


FIG. 206.—Typical installation of shell-type burner in storage-type hot-water heater. (*Courtesy of Automatic Burner Corporation.*)

Vaporizing burners of this type must be designed and installed so that the entire mechanism may be readily disassembled for cleaning. Over a period of time soot may form on the burners and in the stove, so that periodic cleaning may possibly be required, unless the oil burned be of correct specifications.

Applications of blue-flame vaporizing burners are extensive and varied. Cooking stoves, ranges and ovens constitute a large field of use. Space heaters of various designs—for heating small homes, several rooms of a larger building or fairly large open

spaces, such as private garages or isolated offices—constitute the next most important group. Large numbers of hot-water heaters employ these burners.

It is quite essential that the right grade of oil be burned if satisfaction is to be obtained. A very good grade of No. 1 oil

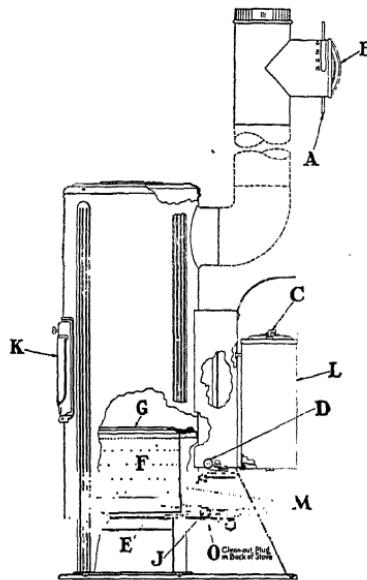


FIG. 207.

FIG. 207.—Sectional view of room heater. *A* and *B*, barometric draft regulator; *C*, oil-tank cover; *D*, constant-level valve adjustment; *E*, oil inlet to burner; *F*, shell of burner; *G*, burner-cover plate; *J*, regulating valve; *K*, heater door; *L*, oil tank; *M*, constant-level float arm; *O*, oil clean-out plug.

FIG. 208.—General appearance of room heater with shell-type oil burner. (*Courtesy of Quaker Manufacturing Company.*)

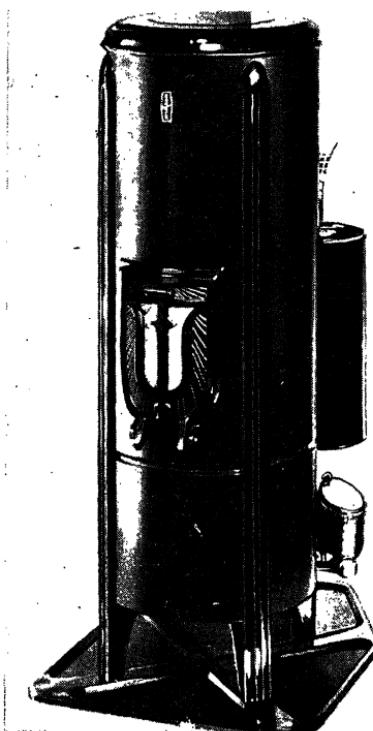


FIG. 208.

may do, providing the flash point is between 110 and 120°F., with an end point around 540°F. Oil having a flash point lower than 110°F. is not safe and should not be used. The oil must be clean and free from sediment and water. Kerosene also is an excellent fuel.

The burning capacity of a single burner ranges from $\frac{1}{2}$ pt. per hr. on small fire for a 6-in. burner up to 3 pt. per hr. maximum for

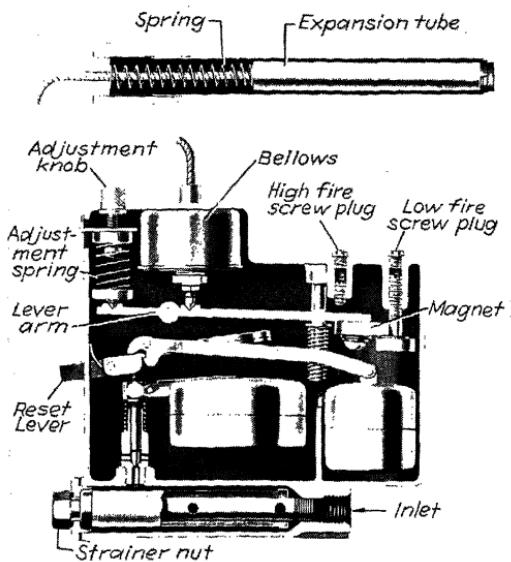


FIG. 209.—Construction of combination constant-level control for gravity feed and thermostatic flow regulator, for hot-water heater. (Courtesy of Automatic Products Company.)

a 10-in. burner. As mentioned above, it is quite necessary that the regulating valves afford easy and dependable adjustment to any desired rate between the minimum and maximum.

CHAPTER XX

CONTROL OF THE HEATING SYSTEM

It is the function of the oil burner and boiler to create or generate steam, and logically this should be done in the most efficient manner possible. But in the heating of major-sized buildings, another factor becomes almost as important as that of efficient steam generation, and that is efficient steam utilization. Properly, the problems of heating plants are not strictly problems of oil-burner operation. Heating-plant problems are identical regardless of the fuel used to generate the steam, or even if exhaust or purchased steam be used. Nevertheless, the problem of heating-plant control becomes inseparable from that of oil-burner control whenever oil burners are used to supply steam primarily to the heating system. It is the purpose of this chapter to consider the relation of burner control to heating-plant control, so that the oil-heated building may be warmed with the least possible fuel consumption.

All heating plants must be designed to meet the most severe weather condition generally prevailing in any particular locality, and are sized to heat a building to 70°F. when the outdoor temperature is at the coldest temperature assumed for design purposes. In most regions this coldest temperature exists less than 5 per cent of the heating season, so that even during the daytime the heating system is not called upon for its maximum output during about 95 per cent of the season. In addition, most buildings do not require 70°F. inside temperature during the night so that here again the load condition is considerably less than the plant's capacity. Therefore it becomes necessary to devise means of controlling the heating-plant output. This can be accomplished manually or automatically, and it can be based upon direct burner control or principally upon control of the heating system itself.

In a great many medium-sized buildings, such as apartments and small hotels, the oil burner can be, and is, operated directly

from a room thermostat located in a position representing as closely as possible an average temperature condition for the entire building. This results in intermittent operation of the burner, and if the heating system is in fair condition and the steam circulation is good, the control will be fair; *i.e.*, the control will be as satisfactory as the thermostat location is truly average for the building. In addition, the fuel consumption will be held within reasonable limits. This method, however, has the objec-

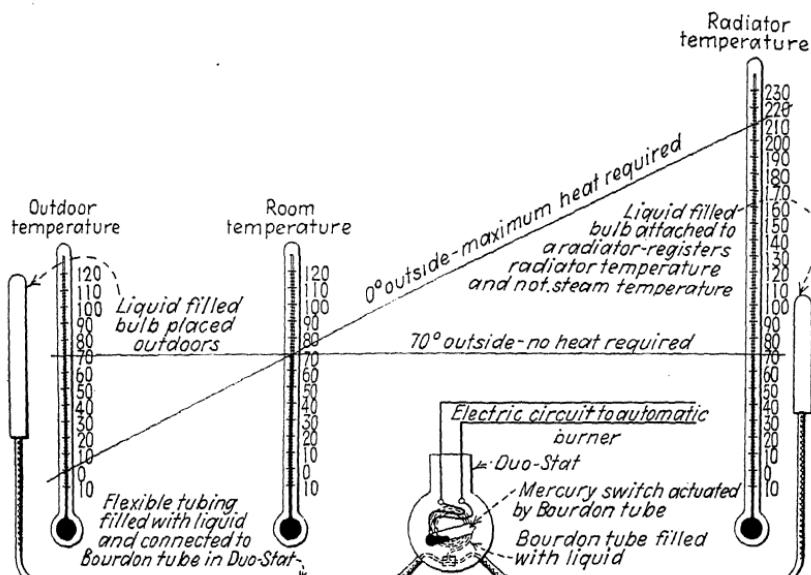


FIG. 210.—Method for determining required radiator temperature for any outdoor temperature on a steam-heating system. (*Courtesy of F. I. Raymond Company.*)

tionable features that it is difficult to pick out a point of average temperature for the thermostat; that the building temperature may vary through a fairly wide range between cycles of burner operation;¹ and that the occupants of the room selected for the thermostat location may, consciously or unconsciously, affect the heating of the entire building.

If the thermostat be omitted entirely, and the burner be placed under control of a pressure switch, then the boiler pressure will be maintained between the upper and the lower limits of the pressure

¹ Refer to the discussion of heat-anticipating thermostat in Chap. X.

switch. This will assure a continuous supply of steam to the radiation, but the only way that the building occupants can regulate room temperature is by opening and closing the radiator valves or windows. This is decidedly an uncertain method of regulation, and the resultant fuel consumption is usually very high. A similar method is to place the burner under modulating or high-low control, so that the fire size, or oil-firing rate, is proportioned to the boiler pressure. Both these methods are extensively used in small- and medium-sized buildings, but as a rule they are wasteful of fuel.

The oil burner can be relieved of direct responsibility for coördinating its operation to heating-plant steam requirements by placing the steam admission to the piping system under the control of a main steam valve. Then the burner has only to maintain a head of steam, so that at any time that the main valve opens steam will be available. In this way the operation of the main control valve would indirectly control the burner, for as soon as the valve either stopped or restricted the flow of steam into the piping system, the burner would build up steam pressure in the boiler, and the burner pressure control then would either cut off or diminish the fire correspondingly. There are a number of commercial forms of apparatus for effecting the operation of such steam valves. Although some of them are adaptable only to steam-valve control, most of them can be connected directly to the burner and accordingly regulate both burner and steam-valve operation simultaneously.

The methods used by the different manufacturers for controlling the supply of steam to the heating system differ considerably. In one system the main steam valve is operated by a control device which is actuated by the outdoor temperature. The lower the outdoor temperature, the shorter the interval between successive openings of the valve. Also the valve will remain open for a greater time when it does open.

In another system, a control device, installed on the roof of the building, is heated electrically at frequent intervals. This control device is so designed that the rate at which it loses heat when the building current is off will match the rate at which the building loses heat. In this way, the time required for the control device to cool, when the heating current is off, determines the admission of steam to the heating system.

A third system uses a number of thermostats in different, representative locations throughout the building. The readings of all these thermostats are averaged by an electrical method. This average temperature of the building then automatically operates the main steam valve controlling the steam supply to the heating system.

It must be borne in mind that there are many factors besides the outside temperature that influence the rate of heat loss from a building. Some of these are wind direction, wind velocity and solar radiation. There are also many factors besides the outside temperature that influence the rate at which heat must be supplied to a building. Among these are relative humidity, individual opinion as to comfortable temperature, percentage of occupancy, hours of building tenancy, ventilation or air changes within the building, etc. Moreover, all types of controls are not alike in their reaction to the different factors; some types will compensate more effectively for certain factors than will other types. Hence it is obvious that building temperature control is a complex subject. The following pages present a review of some commercial types of control equipment, together with an analysis of their operating features, particularly as applied to oil-heated buildings.

CONTROL SYSTEMS HAVING OUTDOOR ELEMENTS

The Raymond Duo-stat.—This device balances outside temperature against radiator temperature, roughly in a manner that will result in a 2° temperature increase above 70°F . in the radiator for each 1° temperature drop below 70°F . outdoors. The instrument consists essentially of a Bourdon tube which actuates a mercury tube for electrical switching. Two fluid-containing bulbs are connected by fine tubing to the Bourdon tube. One bulb is placed outdoors; the other bulb is clamped onto the control radiator in such manner that it registers or responds to the average temperature of the entire radiator. As the outside bulb cools its fluid contracts, causing the Bourdon tube to alter its shape and to tilt the mercury tube into the position which will close an electrical circuit for opening the main steam valve, or starting the stoker, oil burner, or gas burner—as the case may be. As soon as steam reaches the radiator the fluid in the radiator bulb begins to expand. When the expansion in this bulb is equal to

the contraction in the outdoor bulb the Bourdon tube will again assume its normal position, and the electrical circuit will be either opened or reversed, depending on the control wiring. In either case the steam supply to the radiators will be shut off.

The relation between the rise in radiator temperature and the drop in outdoor temperature depends upon the relative sizes of the outdoor and radiator bulbs. For steam systems, the radiator bulb is made twice as long as the outdoor bulb. This relation will cause the instrument to raise the radiator temperature 2° for

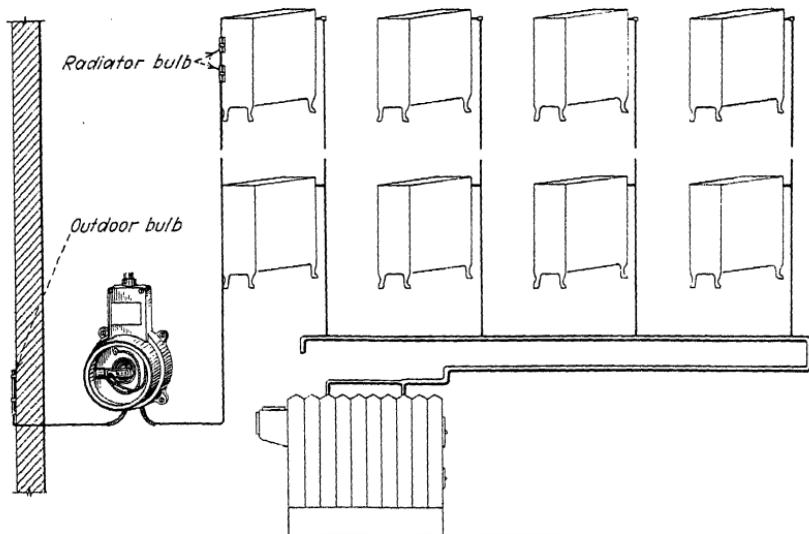


FIG. 211.—Method of connecting Duo-stat to last radiator on system and exposing outdoor bulb.

each 1° that the outdoor temperature drops. If a building is to be heated equally throughout, it is important that the distribution of steam to all the radiators be rapid and uniform. In other words, the first and last radiators should receive steam at approximately the same time. To accomplish this it is frequently necessary to install orifices in the steam supply connection to all of the radiators. This subject of orificing is discussed in some detail later in this chapter.

In addition to the on-and-off type, the Duo-stat can also be supplied in a modifying or throttling type, whereby the steam-control valve is made to assume such a position that steam is supplied constantly and in such quantity that the radiator tem-

perature is held steadily at the point set by the outdoor bulb at any time.

The Marsh Tritrol Regulator.—This device regulates the heating of a building solely from the outside temperature, intermittently operating the burner or steam valve. The frequency and length of the operating periods depend on the outdoor temperature. This instrument is based upon the assumption that the heating plant should operate continuously when the outdoor

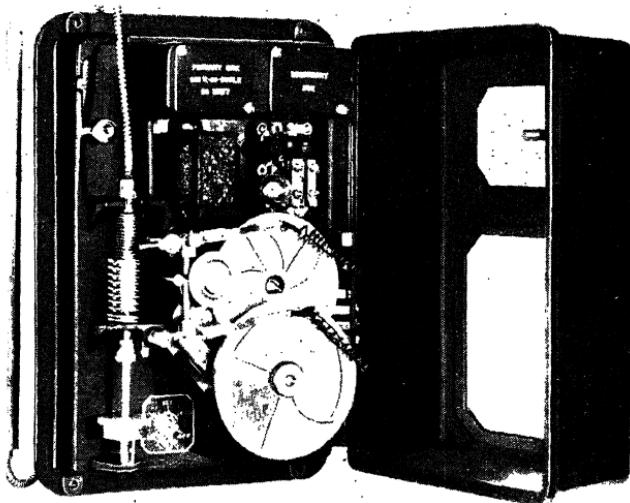


FIG. 212.—General appearance of Marsh Tritrol Regulator.

temperature is as low as the base temperature assumed for design purposes. Furthermore, when the outdoor temperature is higher than this design minimum, the heating system should operate for a correspondingly shorter time. In addition to varying the period and frequency of operation of the firing equipment or the main steam valve, this device also varies the time at which the heating system is turned on in the morning and shut off at night—the colder the weather the earlier the start, and the later the shutoff.

The expansion or contraction of the fluid in the outside bulb causes the selection of the proper path of contact of a contact point upon the surface of a rotating disk, which disk has metallic embossments upon it to complete an electrical circuit whenever

the contact point is in a position touching them. The resultant circuit energizes a relay, from which proper switching is effected to control the oil burner or other firing mechanism. The program of operation as well as the frequency of operation may be varied to suit the characteristics of the heating system.

The Minneapolis-Honeywell Weatherstat.—This control device, which is installed on the building roof, consists primarily of a hollow mass of iron inside which an electric heating element and a thermostat are installed. This electric element heats the iron intermittently under control of the thermostat. The rate at which the mass of iron cools depends on the outdoor temperature, wind, velocity, etc. The faster it cools, the greater the frequency with which the heating element is turned on; also the greater the length of time that will be needed to heat the iron when the heating element is turned on.

The control wiring of this device consists of two parallel electrical circuits. When the thermostat, inside the mass of iron, calls for heat, one circuit turns on the heating element in the iron. The other circuit opens the main steam valve to the heating system of the building or starts the boiler-firing device. When the thermostat inside the mass of iron is satisfied, both the electric heating circuit and the circuit of the building heating system are opened. In this way the operation of the building heating system parallels the operation of the electric heater in the mass of iron on the roof. The satisfactory operation of this system depends on how successfully the rate of heat loss from the iron mass follows the rate of heat loss from the building itself. As a rule, this can be satisfactorily adjusted.

The Radiostat.—It is the intent of this system to secure a balance between the outside temperature and the periods during which steam is supplied to a building by balancing three temperatures: outdoors, boiler water and condensate return. The boiler-water temperature is of course used as an index of steam pressure. The three temperature-registering elements are really thermocouples, in which very minute electrical currents are induced just as in a millivoltmeter pyrometer. All three currents are conducted into a master control box where the currents are magnified many hundreds of times by means of electronic tubes. In the control box a sort of Wheatstone bridge balances these three currents into an equilibrium.

When the outdoor temperature is 65°F. and the return pipe is at room temperature, the boiler water is maintained just below the boiling point, which would be around 210°F. in an atmospheric heating system or at some lower temperature in a vacuum system. The boiler-water element circuit is set beforehand for whatever has been determined upon as the lowest boiler pressure to be considered as a minimum. When the outdoor temperature falls, the electrical balance in the master control is upset, and a relay is actuated to start the firing. Immediately, a head of steam is produced (the steam pressure developed being influenced by the outdoor temperature) and is sustained until condensate reaches the farthest point in the return piping and affects the return temperature element. The firing is then interrupted in accordance with that return temperature, or in other words the maintained boiler-water temperature is reduced by an amount indicated by the temperature of the return water at the farthest point from the boiler. The result is that in a vacuum system the differential and therefore steam circulation is regulated in accordance with outdoor temperature.

The most distinctive feature of this system is that the settings of the three device elements are balanced against each other in an entirely empirical manner, gauged solely by the characteristics of each separate heating system where installed. Full flexibility is provided so that adequate adjustment of each factor may be made with extreme accuracy, as may be dictated by the nature and peculiarities of a particular heating plant.

This control equipment can also be installed so as to regulate a main steam valve instead of a stoker or oil burner. The switch governing change to night control is manually operated, so that an attendant can throw the control from day temperature to night temperature, and the reverse, at any time desired. Or a time clock may be used for the same purpose.

It is perhaps worth noting that of the four control devices just described, Tritrol and Weatherstat are based upon the assumption that the rate of steam generation will be constant during the entire period that heat is called for or during which the burner or stoker is supposed to be on. The Duo-stat and Radiostat, on the contrary, do not anticipate a rate of steam generation.

All four of the above systems of control can be used either for direct operation of the oil burner or for controlling a master steam

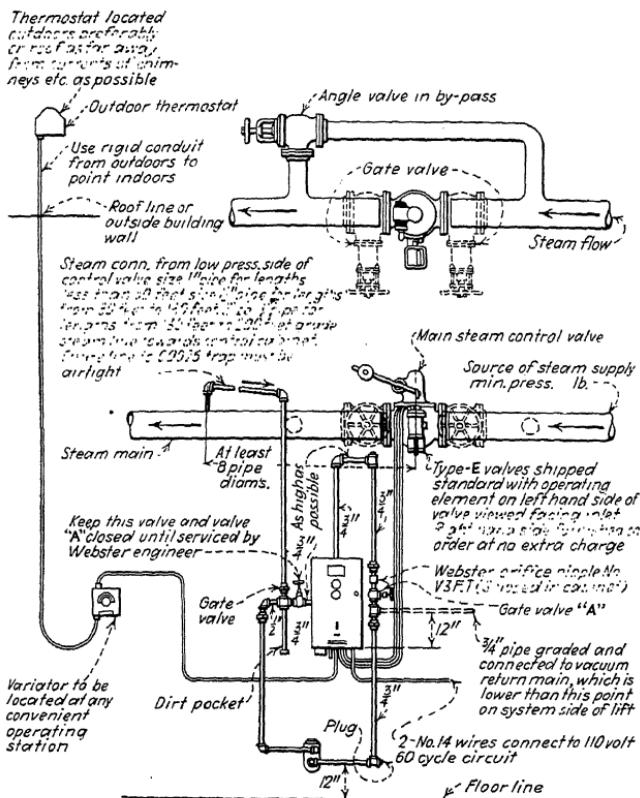
valve. In the latter event the steam valve may regulate steam admission to an entire building, or to one zone of a building. When zoning is employed, each responds individually to its own control instrument. There are also steam-control systems that operate exclusively upon the heating system, leaving the oil burner or other firing device entirely under control of a pressure-stat on the boiler. Typical systems of this nature are briefly described below.

The Dunham Differential System.—Steam admission and steam distribution are the two fundamentals of the Dunham differential vacuum heating system. Control of steam admission is accomplished by a balanced steam valve in the steam main. This valve may be located either at the boiler or at the entrance to a zone. This valve responds to impulses from various sources; a heat balancer, room thermostats and a panel board. Steam distribution is secured by: (1) a vacuum pump which can develop vacuums throughout the system up to 25 in. and (2) by orificing of the radiators. The heat balancer consists of a thermometer bulb inserted into a control radiator and so connected as to send electric signals to the balancer panel, in response to temperature changes within the radiator. The vacuum developed by the pump is regulated from a differential controller, which is actuated by the difference in pressure between the steam supply main and the return main. The thermostats are distributed throughout the building at strategic locations. The readings of all these thermostats are averaged so that as each thermostat calls for heat it causes the opening of a balanced steam valve by a uniform increment.

The Warren Webster Moderator System.—Steam admission and steam distribution are controlled in this system by means of four elements; an outside thermostat called a roof thermostat, a manually manipulated compensator termed a variator, a central control panel which also contains a mercury device for maintaining uniform differential between steam and return lines and a balanced steam valve placed at the entrance to the building heating system or to a zone of a heating system. The moderator system may be operated either electrically or pneumatically.

In the electrical type, the roof thermostat consists of a bimetal coil enclosed within a weatherproof housing on the roof of the building, together with a series of electrical contacts. Changes

in outside temperature cause the bimetal coil to transmit motion to a sliding contact which in turn sends electrical impulses down through corresponding contacts which communicate into the control cabinet in series with the variator. When the variator is in normal position the roof thermostat will cause the steam valve to assume a position proportional to the severity of the



In the pneumatic type, the operation is essentially the same, except that compressed air is the medium used to actuate the system instead of electricity. The roof thermostat is built up of a steel tube about .6 ft. long, inside which is a brass tube fastened at the top to the steel tube. The lower end of the brass tube is free to move, and the motion, which occurs when temperature changes affect it, is transmitted to an air valve which in turn controls operation of the rest of the system as above described for electrical operation. The variator in either case need not be located in the boiler room but can be placed at the most convenient point for the attendant in charge of the heating plant.

As above described the moderator is based upon continuous steam flow, modulated by the position of the steam valve. The system is also made in a form which will give pulsating action by periodic timing of the opening and closing of the steam valve, in short and frequent intervals. The pneumatic type is also available in a simplified form for manual setting of a counterweight on the diaphragm mechanism, the position of the weight being selected according to the severity of the weather. This diaphragm mechanism can then be utilized either for control of the steam valve, or for direct connection to the electrical circuit of the oil burner or stoker.

Illinois Thermo-manual Control System.—This is an automatically controlled steam-circulating system. The control is accomplished by means of a floating type of valve located in either the steam main at the source of supply or in the branch mains at the entrance to each zoned circuit. This valve is responsive to the varying requirements of one or more infiltration thermostats and controls the flow of steam through a full range of heating pressures from about 3 lb. down as low as 24 in. of vacuum. The infiltration thermostat is mounted on the inside face of an outside wall near the return end of a radiator which is normally located at or near the end of the steam circuit. This type of thermostat, owing to its location and design, responds immediately to changes in heating requirements, whether these are due to changes in outside temperatures or wind or sun effect. To maintain comfort conditions within the building or zone this sensitive thermostat determines the position of the floating-type valve to admit the required amount of steam to just balance the

heat losses. A vacuum pump is used to insure rapid and complete circulation of steam. This pump is equipped with a selective controller which insures automatic operation of the pump at the highest vacuum commensurate with the temperature of the returns, and is operative throughout the entire pressure-vacuum range of the system. This system may be operated either electrically or pneumatically, and manual control is provided where required.

RADIATOR ORIFICING

The orificing of radiators serves two functions. In the first place, uniform distribution throughout the entire building, or zone, is achieved. The rush of steam into near radiators is avoided and thus the far radiators can receive steam at about the same time. Secondly, orifices are so designed that the radiator is completely filled with steam only at the one pressure differential (between supply and return mains) upon which the orifice size has been based. When the pressure differential is held at some point less than this maximum value, the orifice will only permit enough steam to flow into the radiator to partially fill it. In this way the heat output of a heating system can be varied by changing the pressure differential across the orifices, even though the flow of steam is continuous. In selecting the orifice diameter, it is necessary to consider the size of the radiator, the operating differential and the distance of any particular radiator from the steam source.

Orificing has also been adapted, in recent years, to one-pipe gravity systems. In this case the operation or flow is intermittent. During the period that steam is flowing, condensate collects in the radiator. While steam is shut off, condensate runs back through the orifice and returns to the boiler. For successful results, the on-and-off periods should be short and of frequent occurrence.

CORRECTION OF PIPING DEFECTS

Intelligent and effective use of a modern temperature-control system is largely dependent upon rapid and uniform steam circulation throughout the system. Any defects in a heating system which tend to retard or obstruct the flow of steam will promptly cause heating difficulties and give rise to complaints from building

occupants. Oddly enough one of the most flagrant and common causes of poor circulation is the one which is easiest to correct. This is improper slope, or pitch, in the radiator runout pipes which connect the radiator to the steam and return risers. Sometimes such a condition arises from poor steam fitting, but more commonly it arises sometime after the installation of the heating plant has been completed, being caused by settling of the building, by changes in radiator location from the original position, etc. Other piping defects encountered are trapped conditions in mains created by improper pitch, by pockets due to pipe dips, by poor or defective dripping of risers and the ends of mains and by sluggish removal of air from the system. In practically every application of control to a heating system some defective conditions will be found to exist in the heating system. Satisfaction can be obtained only by locating and eliminating these defects.

FUEL CONSUMPTION ESTIMATES AND COMPARISONS

Although heating is a branch of engineering, it is very difficult indeed to subject the economics of heating to the same mathematical analysis which is possible in many other branches of engineering—at least, not within any acceptable degree of accuracy. For one thing, reliable data on heating costs are not abundant. For another, such data as are available cannot be rigidly analyzed because of the manifold complications which enter into a tabulation of the factors influencing heating costs and fuel consumption. Finally, fuel in unknown quantities is consumed in most buildings for other purposes, especially hot-water heating. Usually an exact division cannot be made between the various uses for which fuel is expended.

Such attempts as have been made to set up a scale of comparison for fuel consumption for building heating are based upon fuel used per unit of radiation, or unit of building volume, or unit building heat loss per degree day, or per one thousand degree days. The degree day is usually conceived of as a temperature deficiency of 1° through which a building must be heated for 24 hr. when the outside temperature is below 65°F . To allow for reduced night temperature, it is customary to compute degree days by deducting from 65 the average outside temperature for some period of time, usually a month, and to multiply the difference by the number of days in the period of time. Thus, if in

some particular locality the mean outside temperature for the month of January was 23.7°F., then the degree days for that month and locality would be $(65.0 - 23.7) \times 31 = 1280.3$. Variations of this are employed, using other base temperatures as for industrial buildings where the maintained daytime temperature is less than 70°F. The degree day is widely used in the heating profession and offers the most satisfactory basis for expressing heating-plant load conditions so far developed, but as yet, it is doubtful whether the method has been established as a standard conforming to the accuracy of other engineering or scientific standards.

For any one building, it is probable that satisfactory comparisons of fuel consumption can be made for different periods of time, such as different heating seasons or different months, by employing the degree day method. Even then it will generally be found that unit fuel or steam consumption, per degree day, will vary with the time of year, severity of weather, variation of climatic conditions, etc., assuming that all other factors, such as building tenancy and hours of heating, are held constant. However, when this method is used to compare the fuel or steam consumption of different buildings, especially in widely separated localities, there is opportunity for large discrepancies to enter into fuel comparisons. A more (or less) conservative heat loss may have been allowed for in the designs, the building engineers may have divergent opinions and methods in firing and operating the heating-plant apparatus, the percentage and character of building tenancy and the hours of heating may vary, the climatic conditions will be dissimilar and numerous other factors will be at variance with each other—when the fuel consumption for heating one building is compared with that of a supposedly comparable building in another locality.

When analysis is attempted of comparative fuel consumption data, at least the following factors should be considered with respect to their influence upon the heating of the building, and the effectiveness of fuel utilization:

- Load in square feet of radiation or B.t.u. loss from building.
- Degree days for the period and locality under consideration.
- Base temperature used in design of the heating system.
- Daily hours of heating and number of days per week.
- Maintained inside temperature.

Percentage of building occupancy and character of tenancy and uses for which building is employed.

Amount of ventilation or frequency of air changes.

Combustion efficiency for the boiler plant.

Nature and condition of the heating-plant control.

Economy in the operation of the heating plant can be achieved by changes which are divided broadly into two classes:

1. Those which can be achieved by changes in operating methods and maintenance of the heating plant.
2. Those which can be achieved in either the heating plant or the building construction.

Changes in the first class are usually easy to make. They only call for some thought and alertness on the part of the building operator.

Although there are many changes in the second class which it would be impractical to make because of the investment required, there are nevertheless many which are more than justified by the savings incurred. Unless there are prohibitive difficulties because of building construction, possible changes which fall in the second class should be carefully analyzed before being definitely rejected.

ECONOMIES TO BE EFFECTED THROUGH OPERATION AND MAINTENANCE OF THE HEATING PLANT

1. Steam should reach all radiators at approximately the same time. If steam must be supplied for unnecessarily long periods of time in order to heat a few sluggish radiators, fuel will be wasted. Sluggish heating of radiators is due to either trapped supply or return branches, or to air binding.

2. It is important that leaking traps and valves be replaced.

3. Water storage tanks are often installed on the roofs of buildings to provide water for fire protection systems. Steam heaters are usually provided to prevent freezing in cold weather. There is no need for heating these tanks in mild weather. Steam should be turned into these tanks only in freezing weather.

4. Supply heat only when needed. Do not supply heat nights, week-ends or holidays, supply only enough heat to obviate difficulties in building equipment.

5. Shut off radiators in unoccupied portions of the building. During freezing weather turn on only enough radiators to keep the room temperature above freezing.

6. Do not supply heat steadily all day long. Provide heat at intermittent periods. The milder the weather, the longer the off periods can be. This can be accomplished manually, but automatic operation is more satisfactory.

7. Keep the temperature of building as low as is consistent for comfort. Though temperatures of 80° may be required in a hospital, temperatures higher than 72° are seldom justifiable in offices or homes. When a few tenants insist on hothouse temperatures, it is better to install oversized radiators for them than to overheat the entire building.

Temperatures of 50 to 60° are satisfactory for spaces in which men are engaged in active work—not in sedentary bench work. This applies to many shops and garages.

8. Most ventilating systems can deliver far more air than is actually needed. Reduce your air supply to the quantity actually needed. A large amount of fuel is needed during cold weather to heat outdoor air up to building temperatures. Large savings in fuel can be made, by reducing the air supply, without sacrificing the comfort of the building occupants insofar as it is affected by the ventilating systems.

ECONOMIES WHICH CAN BE EFFECTED BY CHANGES IN HEATING SYSTEMS AND BUILDING CONSTRUCTION

1. All steam piping should be covered. Nothing is gained by heating attics or basements.

2. A sheet of insulating material behind radiators under windows prevents the loss of heat to the outside through the wall behind the radiator. This is particularly necessary when the radiator is recessed. The wall behind radiator recesses is usually thin.

3. Radiators should not be covered with the totally enclosed, ornate cabinets so often seen, even when the radiator has been oversized in order to offset the reduced heating capacity of the radiator through the cover. The circulation of the air is affected to such an extent by the cover that steam must be supplied for excessively long periods to heat rooms having radiators covered in this manner.

4. Temperature control of the building is always desirable.

5. Hot water, for the faucet supply, when heated by steam should be thermostatically controlled. For ordinary building

service, water should never be supplied above 140°. Very often cooler water than this will be satisfactory.

6. All control equipment and valves, needed for the daily operation of the plant, should be conveniently located.

7. Orifices installed in the supply valves to all radiators are very beneficial in that all of the radiators will heat uniformly. The last radiator will receive heat as soon as the first one. This eliminates overheating of most of the building in order to heat rooms whose radiators are last on the steam main.

8. Install oversized radiators for the few tenants who insist on the maintenance of tropical 80° temperatures. It is cheaper to do this than it is to carry the whole building at this temperature for a few tenants. The policy of overheating a building for some few tenants will result in open windows all over the building with a resultant waste of heat.

9. Radiators heat rooms more satisfactorily if they are located at outside walls. Best of all is the location of a radiator under the window.

10. Separate steam lines should be installed for the portions of a building which must be heated all night. This is cheaper than heating a whole building in order to provide heat for just a small part of it during the night.

11. Shipping and receiving rooms should be separate from all other rooms. The doors to shipping and receiving rooms should have automatic door closers.

12. Entrances to the building should have vestibules with two sets of doors, or else revolving doors.

13. All windows should be weather-stripped and all window frames caulked.

14. Insulate walls and roof if possible.

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APPENDIX

TABLE XLVI.—TEMPERATURES, FAHRENHEIT AND CENTIGRADE

F.	C.	F.	C.	F.	C.	F.	C.	F.	C.	F.	C.	F.	C.
-40	-40	26	-3.3	92	33.3	158	70	224	106.7	290	143.3	360	182.2
-39	-39.4	27	-2.8	93	33.9	159	70.6	225	107.2	291	143.9	370	187.8
-38	-38.9	28	-2.2	94	34.4	160	71.1	226	107.8	292	144.4	380	193.3
-37	-38.3	29	-1.7	95	35	161	71.7	227	108.3	293	145	390	198.9
-36	-37.8	30	-1.1	96	35.6	162	72.2	228	108.9	294	145.6	400	204.4
-35	-37.2	31	-0.6	97	36.1	163	72.8	229	109.4	295	146.1	410	210
-34	-36.7	32	0	98	36.7	164	73.3	230	110	296	146.7	420	215.6
-33	-36.1	33	+0.6	99	37.2	165	73.9	231	110.6	297	147.2	430	221.1
-32	-35.6	34	1.1	100	37.8	166	74.4	232	111.1	298	147.8	440	226.7
-31	-35	35	1.7	101	38.3	167	75	233	111.7	299	148.3	450	232.2
-30	-34.4	36	2.2	102	38.9	168	75.6	234	112.2	300	148.9	460	237.8
-29	-33.9	37	2.8	103	39.4	169	76.1	235	112.8	301	149.4	470	243.3
-28	-33.3	38	3.3	104	40	170	76.7	236	113.3	302	150	480	248.9
-27	-32.8	39	3.9	105	40.6	171	77.2	237	113.9	303	150.6	490	254.4
-26	-32.2	40	4.4	106	41.1	172	77.8	238	114.4	304	151.1	500	260
-25	-31.7	41	5	107	41.7	173	78.3	239	115	305	151.7	510	265.6
-24	-31.1	42	5.6	108	42.2	174	78.9	240	115.6	306	152.2	520	271.1
-23	-30.6	43	6.1	109	42.8	175	79.4	241	116.1	307	152.8	530	276.7
-22	-30	44	6.7	110	43.3	176	80	242	116.7	308	153.3	540	282.2
-21	-29.4	45	7.2	111	43.9	177	80.6	243	117.2	309	153.9	550	287.8
-20	-28.9	46	7.8	112	44.4	178	81.1	244	117.8	310	154.4	560	293.3
-19	-28.3	47	8.3	113	45	179	81.7	245	118.3	311	155	570	298.9
-18	-27.8	48	8.9	114	45.6	180	82.2	246	118.9	312	155.6	580	304.4
-17	-27.2	49	9.4	115	46.1	181	82.8	247	119.4	313	156.1	590	310
-16	-26.7	50	10	116	46.7	182	83.3	248	120	314	156.7	600	315.6
-15	-26.1	51	10.6	117	47.2	183	83.9	249	120.6	315	157.2	610	321.1
-14	-25.6	52	11.1	118	47.8	184	84.4	250	121.1	316	157.8	620	326.7
-13	-25	53	11.7	119	48.3	185	85	251	121.7	317	158.3	630	332.2
-12	-24.4	54	12	120	48.9	186	85.6	252	122.2	318	158.9	640	337.8
-11	-23.9	55	12.8	121	49.4	187	86.1	253	122.8	319	159.4	650	343.3
-10	-23.3	56	13.3	122	50	188	86.7	254	123.3	320	160	660	348.9
-9	-22.8	57	13.9	123	50.6	189	87.2	255	123.9	321	160.6	670	354.4
-8	-22.2	58	14.4	124	51.1	190	87.8	256	124.4	322	161.1	680	360
-7	-21.7	59	15	125	51.7	191	88.3	257	125	323	161.7	690	365.6
-6	-21.1	60	15.6	126	52.2	192	88.9	258	125.6	324	162.2	700	371.1
-5	-20.6	61	16.1	127	52.8	193	89.4	259	126.1	325	162.8	710	376.7
-4	-20	62	16.7	128	53.3	194	90	260	126.7	326	163.3	720	382.2
-3	-19.4	63	17.2	129	53.9	195	90.6	261	127.2	327	163.9	730	387.8
-2	-18.9	64	17.8	130	54.4	196	91.1	262	127.8	328	164.4	740	393.3
-1	-18.3	65	18.3	131	55	197	91.7	263	128.3	329	165	750	398.9
0	-17.8	66	18.9	132	55.6	198	92.2	264	128.9	330	165.5	760	404.4
+1	-17.2	67	19.4	133	56.1	199	92.8	265	129.4	331	166.1	770	410
2	-16.7	68	20	134	56.7	200	93.3	266	130	332	166.7	780	415.6
3	-16.1	69	20.6	135	57.2	201	93.9	267	130.6	333	167.2	790	421.1
4	-15.6	70	21.1	136	57.8	202	94.4	268	131.1	334	167.8	700	426.7
5	-15	71	21.7	137	58.3	203	95	269	131.7	335	168.3	810	432.2
6	-14.4	72	22	138	58.9	204	95.6	270	132.2	336	168.9	820	437.8
7	-13.9	73	22.8	139	59.4	205	96.1	271	132.8	337	169.4	830	443.3
8	-13.3	74	23.3	140	60	206	96.7	272	133.3	338	170	840	448.9
9	-12.8	75	23.9	141	60.6	207	97.2	273	133.9	339	170.6	850	454.4
10	-12.2	76	24.4	142	61.1	208	97.8	274	134.4	340	171.1	860	460
11	-11.7	77	25	143	61.7	209	98.3	275	135	341	171.7	870	465.6
12	-11.1	78	25.6	144	62.2	210	98.9	276	135.6	342	172.2	880	471.1
13	-10.6	79	26.1	145	62.8	211	99.4	277	136.1	343	172.8	890	476.7
14	-10	80	26.7	146	63.3	212	100	278	136.7	344	173.3	900	482.2
15	-9.4	81	27.2	147	63.9	213	100.6	279	137.2	345	173.9	910	487.8
16	-8.9	82	27.8	148	64.4	214	101.1	280	137.8	346	174.4	920	493.3
17	-8.3	83	28.3	149	65	215	101.7	281	138.3	347	175	930	489.9
18	-7.8	84	28.9	150	65.6	216	102.2	282	138.9	348	175.6	940	504.4
19	-7.2	85	29.4	151	66.1	217	102.8	283	139.4	349	176.1	950	510
20	-6.7	86	30	152	66.7	218	103.3	284	140	350	176.7	960	515.6
21	-6.1	87	30.6	153	67.2	219	103.9	285	140.6	351	177.2	970	521.1
22	-5.6	88	31.1	154	67.8	220	104.4	286	141.1	352	177.8	980	526.7
23	-5	89	31.7	155	68.3	221	105	287	141.7	353	178.3	990	532.2
24	-4.4	90	32.2	156	68.9	222	105.6	288	142.2	354	178.9	1000	537.8
25	-3.9	91	32.8	157	69.4	223	106.1	289	142.8	355	179.4	1010	543.3

OIL BURNERS

TABLE XLVII.—TEMPERATURES, CENTIGRADE AND FAHRENHEIT

C.	F.	C.	F.	C.	F.	C.	F.	C.	F.	C.	F.	C.	F.
-40	-40	26	78.8	92	197.6	158	316.4	224	435.2	290	554	950	1,742
-39	-38.2	27	80.6	93	199.4	159	318.2	225	437	300	572	960	1,760
-38	-36.4	28	82.4	94	201.2	160	320	226	438.8	310	590	970	1,778
-37	-34.6	29	84.2	95	203	161	321.8	227	440.6	320	608	980	1,796
-36	-32.8	30	86	96	204.8	162	323.6	228	442.4	330	626	990	1,814
-35	-31	31	87.8	97	206.6	163	324.4	229	444.2	340	644	1,000	1,832
-34	-29.2	32	89.6	98	208.4	164	327.2	230	446	350	662	1,010	1,850
-33	-27.4	33	91.4	99	210.2	165	329	231	447.8	360	680	1,020	1,868
-32	-25.6	34	93.2	100	212	166	330.8	232	449.6	370	698	1,030	1,886
-31	-23.8	35	95	101	231.8	167	332.6	233	451.4	380	716	1,040	1,904
-30	-22	36	96.8	102	215.6	168	334.4	234	453.2	390	734	1,050	1,922
-29	-20.2	37	98.6	103	217.4	169	336.2	235	455	400	752	1,060	1,940
-28	-18.4	38	100.4	104	219.2	170	338	236	456.8	410	770	1,070	1,958
-27	-16.6	39	102.2	105	221	171	339.8	237	458.6	420	788	1,080	1,976
-26	-14.8	40	104	106	222.8	172	341.6	238	460.4	430	806	1,090	1,994
-25	-13	41	105.8	107	224.6	173	343.4	239	462.2	440	824	1,100	2,012
-24	-11.2	42	107.6	108	226.4	174	345.2	240	464	450	842	1,110	2,030
-23	-9.4	43	109.4	109	228.2	175	347	241	465.8	460	860	1,120	2,048
-22	-7.6	44	111.2	110	230	176	348.8	242	467.6	470	878	1,130	2,066
-21	-5.8	45	113	111	231.8	177	350.6	243	469.4	480	896	1,140	2,084
-20	-4	46	114.8	112	233.6	178	352.4	244	471.2	490	914	1,150	2,102
-19	-2.2	47	116.6	113	235.4	179	354.2	245	473	500	932	1,160	2,120
-18	-0.4	48	118.4	114	237.2	180	356	246	474.8	510	950	1,170	2,138
-17	+1.4	49	120.2	115	239	181	357.8	247	476.6	520	968	1,180	2,156
-16	3.2	50	122	116	240.8	182	359.6	248	478.4	530	986	1,190	2,174
-15	5	51	123.8	117	242.6	183	361.4	249	480.2	540	1,004	1,200	2,192
-14	6.8	52	125.6	118	244.4	184	363.2	250	482	550	1,022	1,210	2,210
-13	8.6	53	127.4	119	246.2	185	365	251	483.8	560	1,040	1,220	2,228
-12	10.4	54	129.2	120	248	186	366.8	252	485.6	570	1,058	1,230	2,246
-11	12.2	55	131	121	249.8	187	368.6	253	487.4	580	1,076	1,240	2,264
-10	14	56	132.8	122	251.6	188	370.4	254	489.2	590	1,094	1,250	2,282
-9	15.8	57	134.6	123	253.4	189	372.2	255	491	600	1,112	1,260	300
-8	17.6	58	136.4	124	255.2	190	374	256	492.8	610	1,130	1,270	318
-7	19.4	59	138.2	125	257	191	375.8	257	494.6	620	1,148	1,280	336
-6	21.2	60	140	126	258.8	192	377.6	258	496.4	630	1,166	1,290	354
-5	23	61	141.8	127	260.6	193	379.4	259	498.2	640	1,184	1,300	2,372
-4	24.8	62	143.6	128	262.4	194	381.2	260	500	650	1,202	1,310	2,390
-3	26.6	63	145.4	129	264.2	195	383	261	501.8	660	1,220	1,320	2,408
-2	28.4	64	147.2	130	266	196	384.8	262	503.6	670	1,238	1,330	2,426
-1	30.2	65	149	131	267.8	197	386.6	263	505.4	680	1,256	1,340	2,444
0	32	66	150.8	132	269.6	198	388.4	264	507.2	690	1,274	1,350	2,462
+1	33.8	67	152.6	133	271.4	199	390.2	265	509	700	1,292	1,360	2,480
2	35.6	68	154.4	134	273.2	200	392	266	510.8	710	1,310	1,370	2,498
3	37.4	69	156.2	135	275	201	393.8	267	512.6	720	1,328	1,380	2,516
4	39.2	70	158	136	276.8	202	395.6	268	514.4	730	1,346	1,390	2,534
5	41	71	159.8	137	278.6	203	397.4	269	516.2	740	1,364	1,400	2,552
6	42.8	72	161.6	138	280.4	204	399.2	270	518	750	1,382	1,410	2,570
7	44.6	73	163.4	139	282.2	205	401	271	519.8	760	1,400	1,420	2,588
8	46.4	74	165.2	140	284	206	402.8	272	521.6	770	1,418	1,430	2,606
9	48.2	75	167	141	285.8	207	404.6	273	523.4	780	1,436	1,440	2,624
10	50	76	168.8	142	287.6	208	406.4	274	525.2	790	1,454	1,450	2,642
11	51.8	77	170.6	143	289.4	209	408.2	275	527	800	1,472	1,460	2,660
12	53.6	78	172.4	144	291.2	210	410	276	528.8	810	1,490	1,470	2,678
13	55.4	79	174.2	145	293	211	411.8	277	530.6	820	1,508	1,480	2,696
14	57.2	80	176	146	294.8	212	413.6	278	532.4	830	1,526	1,490	2,714
15	59	81	177.8	147	296.6	213	415.4	279	534.2	840	1,544	1,500	2,732
16	60.8	82	179.6	148	298.4	214	417.2	280	536	850	1,562	1,510	2,750
17	62.6	83	181.4	149	300.2	215	419	281	537.8	860	1,580	1,520	2,768
18	64.4	84	183.2	150	302	216	420.8	282	539.6	870	1,598	1,530	2,786
19	66.2	85	185	151	303.8	217	422.6	283	541.4	880	1,616	1,540	2,804
20	68	86	186.8	152	305.6	218	424.4	284	543.2	890	1,634	1,550	2,822
21	69.8	87	188.6	153	307.4	219	426.2	285	545	900	1,652	1,600	2,912
22	71.6	88	190.4	154	309.2	220	428	286	546.8	910	1,670	1,650	3,002
23	73.4	89	192.2	155	311	221	429.8	287	548.6	920	1,688	1,700	3,092
24	75.2	90	194	156	312.8	222	431.6	288	550.4	930	1,706	1,750	3,182
25	77	91	195.8	157	314.6	223	433.4	289	552.2	940	1,724	1,800	3,272

TABLE XLVIII.—PROPERTIES OF SATURATED STEAM (BRITISH UNITS)

Absolute pressure, lb. per sq. in.	Temp., °F.	Specific volume, cu. ft. per lb.	Density, lb. per cu. ft.	Heat of liquid above 32°F.	Latent heat of evaporation	Total heat of steam above 32°F.	Internal energy (B.t.u.), evaporation	Entropy	
								Water	Evaporation
1	101.83	333.0	0.00300	69.8	1,034.6	1,104.4	972.9	0.1327	1.8427
5	162.28	73.33	0.01364	130.1	1,000.3	1,130.5	932.4	0.2348	1.6084
10	193.22	38.38	0.02606	161.1	982.0	1,143.1	910.9	0.2832	1.5042
14.7	212.00	26.79	0.03732	180.0	970.4	1,150.4	897.6	0.3118	1.4447
15	213.00	26.27	0.03806	181.0	969.7	1,150.7	896.8	0.3133	1.4416
20	228.00	20.08	0.04980	196.1	960.0	1,156.2	885.8	0.3355	1.3965
25	240.1	16.30	0.0614	208.4	952.0	1,160.4	876.8	0.3532	1.3604
30	250.3	13.74	0.0728	218.8	945.1	1,163.9	869.0	0.3680	1.3311
35	259.3	11.89	0.0841	227.9	938.9	1,168.8	862.1	0.3808	1.3060
40	267.3	10.49	0.0953	236.1	933.3	1,169.4	855.9	0.3920	1.2841
45	274.5	9.39	0.1065	243.4	928.2	1,171.6	850.3	0.4021	1.2644
50	281.0	8.51	0.1175	250.1	923.5	1,173.6	845.0	0.4113	1.2468
55	287.1	7.78	0.1285	256.3	919.0	1,175.4	840.2	0.4196	1.2309
60	292.7	7.17	0.1394	262.1	914.9	1,177.0	835.6	0.4272	1.2160
65	298.0	6.65	0.1503	267.5	911.0	1,178.5	831.4	0.4344	1.2024
70	302.9	6.20	0.1612	272.6	907.2	1,179.8	827.3	0.4411	1.1896
75	307.6	5.81	0.1721	277.4	903.7	1,181.1	823.5	0.4474	1.1778
80	312.0	5.47	0.1829	282.0	900.3	1,182.3	819.8	0.4535	1.1665
85	316.3	5.16	0.1937	286.3	897.1	1,183.4	816.3	0.4590	1.1561
90	320.3	4.89	0.2044	290.5	893.9	1,184.4	813.0	0.4644	1.1461
95	324.1	4.65	0.2151	294.5	890.9	1,185.4	809.7	0.4694	1.1367
100	327.8	4.429	0.2258	298.3	888.0	1,186.3	806.6	0.4743	1.1277
105	331.4	4.230	0.2365	302.0	885.2	1,187.2	803.6	0.4789	1.1191
110	334.8	4.047	0.2472	305.5	882.5	1,188.0	800.7	0.4834	1.1108
115	338.1	3.880	0.2577	309.0	879.8	1,188.8	797.9	0.4877	1.1030
120	341.3	3.726	0.2683	312.3	877.2	1,189.6	795.2	0.4919	1.0954
125	344.4	3.583	0.2791	315.5	874.7	1,190.3	792.6	0.4959	1.0880
130	347.4	3.452	0.2897	318.6	872.3	1,191.0	790.0	0.4998	1.0809
135	350.3	3.331	0.3002	321.7	869.9	1,191.6	787.5	0.5035	1.0742
140	353.1	3.219	0.3107	324.6	867.6	1,192.2	785.0	0.5072	1.0675
145	355.8	3.112	0.3213	327.4	865.4	1,192.8	782.7	0.5107	1.0621
150	358.5	3.012	0.3320	330.2	863.2	1,193.4	780.4	0.5142	1.0550
155	361.0	2.920	0.3425	332.9	861.0	1,194.0	778.1	0.5175	1.0489
160	363.6	2.834	0.3529	335.6	858.8	1,194.5	775.8	0.5208	1.0431
165	366.0	2.753	0.3633	338.2	856.8	1,195.0	773.6	0.5239	1.0376
170	368.5	2.675	0.3738	340.7	854.7	1,195.4	771.5	0.5269	1.0321
175	370.8	2.602	0.3843	343.2	852.7	1,195.9	769.4	0.5299	1.0268
180	373.1	2.533	0.3948	345.6	850.8	1,196.4	767.4	0.5328	1.0215
185	375.4	2.468	0.4052	348.0	848.4	1,196.8	765.4	0.5356	1.0164
190	377.6	2.406	0.4157	350.4	846.9	1,197.3	763.4	0.5384	1.0114

TABLE XLVIII.—PROPERTIES OF SATURATED STEAM (BRITISH UNITS).—
(Continued)

Absolute pressure, lb. per sq. in.	Temp., °F.	Specific volume, cu. ft. per lb.	Density, lb. per cu. ft.	Heat of liquid above 32°F.	Latent heat of evaporation	Total heat of steam above 32°F.	Internal energy (B.t.u.), evaporation	Entropy	
								Water	Evaporation
195	379.8	2.346	0.4262	352.7	845.0	1,197.7	761.4	0.5410	1.0066
200	381.9	2.290	0.437	354.9	843.2	1,198.1	759.5	0.5437	1.0019
205	384.0	2.237	0.447	357.1	841.4	1,198.5	757.6	0.5463	0.9973
210	386.0	2.187	0.457	359.2	839.6	1,198.8	755.8	0.5488	0.9928
215	388.0	2.138	0.468	361.4	837.9	1,199.2	754.0	0.5513	0.9885
220	389.9	2.091	0.478	363.4	836.2	1,199.6	752.3	0.5538	0.9841
225	391.9	2.046	0.489	365.5	834.4	1,199.9	750.5	0.5562	0.9799
230	393.8	2.004	0.499	367.5	832.1	1,200.2	748.8	0.5586	0.9758
235	395.6	1.964	0.509	369.4	831.1	1,200.6	747.0	0.5610	0.9717
240	397.4	1.924	0.520	371.4	829.5	1,200.9	745.4	0.5633	0.9676
245	399.3	1.887	0.530	373.3	827.9	1,201.2	743.7	0.5655	0.9638
250	401.1	1.850	0.541	375.2	826.3	1,201.5	742.0	0.5676	0.9600
260	404.1	1.782	0.561	378.9	823.1	1,202.1	738.9	0.5719	0.9525
270	407.9	1.718	0.582	382.5	820.1	1,202.6	735.8	0.5760	0.9454
280	411.2	1.658	0.603	386.1	817.1	1,203.1	732.7	0.5800	0.9385
290	414.4	1.602	0.624	389.4	814.2	1,203.6	729.7	0.5840	0.9316
300	417.5	1.551	0.645	392.7	811.3	1,204.1	726.8	0.5878	0.9251
310	420.5	1.502	0.666	395.9	808.5	1,204.5	724.0	0.5915	0.9187
320	423.4	1.456	0.687	399.1	805.8	1,204.9	721.2	0.5951	0.9125
330	426.3	1.413	0.708	402.2	803.1	1,205.3	718.5	0.5986	0.9065
340	429.1	1.372	0.729	405.3	800.4	1,205.7	715.9	0.6020	0.9006
350	431.9	1.334	0.750	408.2	797.8	1,206.1	713.3	0.6053	0.8949
360	434.6	1.298	0.770	411.2	795.3	1,206.4	710.7	0.6085	0.8894
370	437.2	1.264	0.791	414.0	792.8	1,206.8	708.2	0.6116	0.8840
380	439.8	1.231	0.812	416.8	790.3	1,207.1	705.7	0.6147	0.8788
390	442.3	1.200	0.833	419.5	787.9	1,207.4	703.3	0.6178	0.8737
400	444.8	1.17	0.86	422.0	786.0	1,208.0	701.0	0.621	0.868
450	456.5	1.04	0.96	435.0	774.0	1,209.0	690.0	0.635	0.844
500	467.3	0.93	1.08	448.0	762.0	1,210.0	678.0	0.648	0.822
550	477.3	0.83	1.20	459.0	751.0	1,210.0	668.0	0.659	0.803
600	486.6	0.76	1.32	469.0	741.0	1,210.0	658.0	0.670	0.783

The pressures in this table are absolute pressures. Convert the corrected barometer reading into pounds per square inch, add this pressure to the gauge pressure, and with this absolute pressure enter the table for the desired data.

TABLE XLIX.—REDUCTION OF OBSERVED DEGREES A.P.I. TO DEGREES A.P.I. AT 60°F.

Observed temp., °F.	Observed degrees A.P.I.									
	10	11	12	13	14	15	16	17	18	19
	Corresponding degrees A.P.I. at 60°F.									
30	11.5	12.5	13.6	14.6	15.6	16.6	17.7	18.7	19.7	20.7
31	11.5	12.5	13.5	14.5	15.6	16.6	17.6	18.6	19.7	20.7
32	11.4	12.4	13.4	14.5	15.5	16.5	17.6	18.6	19.6	20.6
33	11.4	12.4	13.4	14.4	15.5	16.5	17.5	18.5	19.5	20.6
34	11.3	12.3	13.3	14.4	15.4	16.4	17.4	18.4	19.5	20.5
35	11.3	12.3	13.3	14.3	15.3	16.4	17.4	18.4	19.4	20.4
36	11.2	12.2	13.2	14.3	15.3	16.3	17.3	18.3	19.4	20.4
37	11.2	12.2	13.2	14.2	15.2	16.3	17.3	18.3	19.3	20.3
38	11.1	12.1	13.1	14.2	15.2	16.2	17.2	18.2	19.2	20.3
39	11.1	12.1	13.1	14.1	15.1	16.1	17.2	18.2	19.2	20.2
40	11.0	12.0	13.0	14.0	15.1	16.1	17.1	18.1	19.1	20.1
41	11.0	12.0	13.0	14.0	15.0	16.0	17.0	18.1	19.1	20.1
42	10.9	11.9	12.9	13.9	15.0	16.0	17.0	18.0	19.0	20.0
43	10.8	11.9	12.9	13.9	14.9	15.9	16.9	17.9	18.9	20.0
44	10.8	11.8	12.8	13.8	14.9	15.9	16.9	17.9	18.9	19.9
45	10.7	11.8	12.8	13.8	14.8	15.8	16.8	17.8	18.8	19.9
46	10.7	11.7	12.7	13.7	14.8	15.8	16.8	17.8	18.8	19.8
47	10.6	11.7	12.7	13.7	14.7	15.7	16.7	17.7	18.7	19.7
48	10.6	11.6	12.6	13.6	14.6	15.6	16.7	17.7	18.7	19.7
49	10.6	11.6	12.6	13.6	14.6	15.6	16.6	17.6	18.6	19.6
50	10.5	11.5	12.5	13.5	14.5	15.5	16.5	17.5	18.5	19.6
51	10.5	11.5	12.5	13.5	14.5	15.5	16.5	17.5	18.5	19.5
52	10.4	11.4	12.4	13.4	14.4	15.4	16.4	17.4	18.4	19.4
53	10.4	11.4	12.4	13.4	14.4	15.4	16.4	17.4	18.4	19.4
54	10.3	11.3	12.3	13.3	14.3	15.3	16.3	17.3	18.3	19.3
55	10.3	11.3	12.3	13.3	14.3	15.3	16.3	17.3	18.3	19.3
56	10.2	11.2	12.2	13.2	14.2	15.2	16.2	17.2	18.2	19.2
57	10.1	11.1	12.1	13.1	14.1	15.1	16.2	17.2	18.2	19.2
58	10.1	11.1	12.1	13.1	14.1	15.1	16.1	17.1	18.1	19.1
59	10.0	11.0	12.0	13.0	14.0	15.0	16.0	17.0	18.0	19.0
60	10.0	11.0	12.0	13.0	14.0	15.0	16.0	17.0	18.0	19.0
61	9.9	10.9	11.9	12.9	13.9	14.9	15.9	16.9	17.9	18.9
62	9.9	10.9	11.9	12.9	13.9	14.9	15.9	16.9	17.9	18.9
63	9.9	10.8	11.8	12.8	13.8	14.8	15.8	16.8	17.8	18.8
64	9.8	10.8	11.8	12.8	13.8	14.8	15.8	16.8	17.8	18.8
65	9.8	10.7	11.7	12.7	13.7	14.7	15.7	16.7	17.7	18.7
66	9.7	10.7	11.7	12.7	13.7	14.7	15.7	16.7	17.7	18.7
67	9.7	10.6	11.6	12.6	13.6	14.6	15.6	16.6	17.6	18.6
68	9.6	10.6	11.6	12.6	13.6	14.6	15.6	16.6	17.6	18.5
69	9.6	10.6	11.5	12.5	13.5	14.5	15.5	16.5	17.5	18.5
70	9.5	10.5	11.5	12.5	13.5	14.5	15.5	16.4	17.4	18.4
71	9.5	10.5	11.4	12.5	13.4	14.4	15.4	16.4	17.4	18.4
72	9.4	10.4	11.4	12.4	13.4	14.4	15.4	16.3	17.3	18.3
73	9.4	10.4	11.3	12.3	13.3	14.3	15.3	16.3	17.3	18.3
74	9.3	10.3	11.3	12.3	13.3	14.3	15.2	16.2	17.2	18.2
75	9.3	10.2	11.2	12.2	13.2	14.2	15.2	16.2	17.2	18.2
76	9.2	10.2	11.2	12.2	13.2	14.2	15.1	16.1	17.1	18.1
77	9.2	10.1	11.1	12.1	13.1	14.1	15.1	16.1	17.1	18.0
78	9.1	10.1	11.1	12.1	13.1	14.0	15.0	16.0	17.0	18.0
79	9.1	10.0	11.0	12.0	13.0	14.0	15.0	16.0	17.0	17.9
80	9.0	10.0	11.0	12.0	13.0	13.9	14.9	15.9	16.9	17.9
81	9.0	9.9	10.9	11.9	12.9	13.9	14.9	15.8	16.8	17.8
82	8.9	9.9	10.9	11.9	12.9	13.8	14.8	15.8	16.8	17.8
83	8.9	9.8	10.8	11.8	12.8	13.8	14.8	15.7	16.7	17.7
84	8.8	9.8	10.8	11.8	12.8	13.7	14.7	15.7	16.7	17.7

OIL BURNERS

TABLE XLIX.—REDUCTION OF OBSERVED DEGREES A.P.I. TO DEGREES A.P.I. AT 60°F.—(Continued)

Observed temp., °F.	Observed degrees A.P.I.									
	10	11	12	13	14	15	16	17	18	19
Corresponding degrees A.P.I. at 60°F.										
85	8.8	9.8	10.7	11.7	12.7	13.7	14.7	15.6	16.6	17.6
86	8.7	9.7	10.7	11.7	12.6	13.6	14.6	15.6	16.6	17.6
87	8.7	9.7	10.6	11.6	12.6	13.6	14.6	15.5	16.5	17.5
88	8.6	9.6	10.6	11.6	12.5	13.5	14.5	15.5	16.5	17.4
89	8.6	9.6	10.5	11.5	12.5	13.5	14.5	15.4	16.4	17.4
90	8.5	9.5	10.5	11.5	12.4	13.4	14.4	15.4	16.4	17.3
91	8.5	9.5	10.4	11.4	12.4	13.4	14.4	15.3	16.3	17.3
92	8.4	9.4	10.4	11.4	12.3	13.3	14.3	15.3	16.3	17.2
93	8.4	9.4	10.3	11.3	12.3	13.3	14.3	15.2	16.2	17.2
94	8.3	9.3	10.3	11.3	12.2	13.2	14.2	15.2	16.2	17.1
95	8.3	9.3	10.2	11.2	12.2	13.2	14.2	15.1	16.1	17.1
96	8.2	9.2	10.2	11.2	12.1	13.1	14.1	15.1	16.0	17.0
97	8.2	9.2	10.1	11.1	12.1	13.1	14.0	15.0	16.0	17.0
98	8.1	9.1	10.1	11.1	12.0	13.0	14.0	15.0	15.9	16.9
99	8.1	9.1	10.0	11.0	12.0	13.0	13.9	14.9	15.9	16.8
100	8.1	9.0	10.0	11.0	11.9	12.9	13.9	14.9	15.8	16.8
101	8.0	9.0	9.9	10.9	11.9	12.9	13.8	14.8	15.8	16.7
102	8.0	8.9	9.9	10.9	11.8	12.8	13.8	14.8	15.7	16.7
103	7.9	8.9	9.9	10.8	11.8	12.8	13.7	14.7	15.7	16.6
104	7.9	8.8	9.8	10.8	11.7	12.7	13.7	14.7	15.6	16.6
105	7.8	8.8	9.8	10.7	11.7	12.7	13.6	14.6	15.6	16.5
106	7.8	8.8	9.7	10.7	11.7	12.6	13.6	14.6	15.5	16.5
107	7.7	8.7	9.7	10.6	11.6	12.6	13.5	14.5	15.5	16.4
108	7.7	8.7	9.6	10.6	11.6	12.5	13.5	14.5	15.4	16.4
109	7.6	8.6	9.6	10.5	11.5	12.5	13.4	14.4	15.4	16.3
110	7.6	8.6	9.5	10.5	11.5	12.4	13.4	14.4	15.3	16.3
111	7.6	8.5	9.5	10.4	11.4	12.4	13.3	14.3	15.3	16.2
112	7.5	8.5	9.4	10.4	11.4	12.3	13.3	14.2	15.2	16.2
113	7.5	8.4	9.4	10.3	11.3	12.3	13.2	14.2	15.1	16.1
114	7.4	8.4	9.3	10.3	11.3	12.2	13.2	14.1	15.1	16.1
115	7.4	8.3	9.3	10.2	11.2	12.2	13.1	14.1	15.0	16.0
116	7.3	8.3	9.2	10.2	11.2	12.1	13.1	14.0	15.0	16.0
117	7.3	8.2	9.2	10.1	11.1	12.1	13.0	14.0	14.9	15.9
118	7.2	8.2	9.1	10.1	11.1	12.0	13.0	13.9	14.9	15.9
119	7.2	8.1	9.1	10.1	11.0	12.0	12.9	13.9	14.8	15.8
Observed degrees A.P.I.										
Observed temp., °F.	20	21	22	23	24	25	26	27	28	29
	Corresponding degrees A.P.I. at 60°F.									
30	21.7	22.8	23.8	24.8	25.8	26.9	27.9	28.9	30.0	31.0
31	21.7	22.7	23.7	24.7	25.8	26.8	27.8	28.9	29.9	30.9
32	21.6	22.7	23.7	24.7	25.7	26.8	27.8	28.8	29.8	30.9
33	21.6	22.6	23.6	24.6	25.7	26.7	27.7	28.7	29.8	30.8
34	21.5	22.5	23.6	24.6	25.6	26.6	27.7	28.7	29.7	30.7
35	21.4	22.5	23.5	24.5	25.5	26.6	27.6	28.6	29.6	30.7
36	21.4	22.4	23.4	24.4	25.5	26.5	27.5	28.5	29.6	30.6
37	21.3	22.3	23.4	24.4	25.4	26.4	27.4	28.5	29.5	30.5
38	21.3	22.3	23.3	24.3	25.3	26.3	27.4	28.4	29.4	30.4
39	21.2	22.2	23.3	24.3	25.3	26.3	27.3	28.3	29.4	30.4

TABLE XLIX.—REDUCTION OF OBSERVED DEGREES A.P.I. TO DEGREES
 A.P.I. AT 60°F.—(Continued)

Observed temp., °F.	Observed degrees A.P.I.									
	20	21	22	23	24	25	26	27	28	29
	Corresponding degrees A.P.I. at 60°F.									
40	21.2	22.2	23.2	24.2	25.2	26.2	27.3	28.3	29.3	30.3
41	21.1	22.1	23.1	24.2	25.2	26.2	27.2	28.2	29.2	30.3
42	21.0	22.1	23.1	24.1	25.1	26.1	27.1	28.2	29.2	30.2
43	21.0	22.0	23.0	24.0	25.0	26.0	27.1	28.1	29.1	30.1
44	20.9	21.9	23.0	24.0	25.0	26.0	27.0	28.0	29.0	30.0
45	20.9	21.9	22.9	23.9	24.9	25.9	26.9	28.0	29.0	30.0
46	20.8	21.8	22.8	23.8	24.8	25.9	26.9	27.9	28.9	29.9
47	20.7	21.7	22.8	23.8	24.8	25.8	26.8	27.8	28.8	29.8
48	20.7	21.7	22.7	23.7	24.7	25.7	26.8	27.8	28.8	29.8
49	20.6	21.6	22.7	23.7	24.7	25.7	26.7	27.7	28.7	29.7
50	20.6	21.6	22.6	23.6	24.6	25.6	26.6	27.6	28.7	29.7
51	20.5	21.5	22.5	23.5	24.5	25.5	26.6	27.6	28.6	29.6
52	20.5	21.5	22.5	23.5	24.5	25.5	26.5	27.5	28.5	29.5
53	20.4	21.4	22.4	23.4	24.4	25.4	26.4	27.4	28.5	29.5
54	20.3	21.3	22.3	23.3	24.4	25.4	26.4	27.4	28.4	29.4
55	20.3	21.3	22.3	23.3	24.3	25.3	26.3	27.3	28.3	29.3
56	20.2	21.2	22.2	23.2	24.2	25.2	26.2	27.3	28.3	29.3
57	20.2	21.2	22.2	23.2	24.2	25.2	26.2	27.2	28.2	29.2
58	20.1	21.1	22.1	23.1	24.1	25.1	26.1	27.1	28.1	29.1
59	20.1	21.1	22.1	23.1	24.1	25.1	26.1	27.1	28.1	29.1
60	20.0	21.0	22.0	23.0	24.0	25.0	26.0	27.0	28.0	29.0
61	19.9	20.9	21.9	22.9	23.9	24.9	25.9	26.9	27.9	28.9
62	19.9	20.9	21.9	22.9	23.9	24.9	25.9	26.9	27.9	28.9
63	19.8	20.8	21.8	22.8	23.8	24.8	25.8	26.8	27.8	28.8
64	19.8	20.8	21.8	22.8	23.8	24.8	25.8	26.8	27.8	28.8
65	19.7	20.7	21.7	22.7	23.7	24.7	25.7	26.7	27.7	28.7
66	19.7	20.7	21.7	22.6	23.6	24.6	25.6	26.6	27.6	28.6
67	19.6	20.6	21.6	22.6	23.6	24.6	25.6	26.6	27.6	28.5
68	19.5	20.5	21.5	22.5	23.5	24.5	25.5	26.5	27.5	28.5
69	19.5	20.5	21.5	22.4	23.4	24.4	25.4	26.4	27.4	28.4
70	19.4	20.4	21.4	22.4	23.4	24.4	25.4	26.4	27.4	28.3
71	19.4	20.4	21.4	22.3	23.3	24.3	25.3	26.3	27.3	28.3
72	19.3	20.3	21.3	22.3	23.3	24.3	25.2	26.2	27.2	28.2
73	19.2	20.2	21.2	22.2	23.2	24.2	25.2	26.2	27.2	28.2
74	19.2	20.2	21.2	22.2	23.1	24.1	25.1	26.1	27.1	28.1
75	19.1	20.1	21.1	22.1	23.1	24.1	25.1	26.0	27.0	28.0
76	19.1	20.1	21.1	22.0	23.0	24.0	25.0	26.0	27.0	28.0
77	19.0	20.0	21.0	22.0	23.0	24.0	25.0	25.9	26.9	27.9
78	19.0	20.0	20.9	21.9	22.9	23.9	24.9	25.9	26.8	27.8
79	18.9	19.9	20.9	21.8	22.8	23.8	24.8	25.8	26.8	27.8
80	18.9	19.9	20.8	21.8	22.8	23.8	24.8	25.7	26.7	27.7
81	18.8	19.8	20.8	21.7	22.7	23.7	24.7	25.7	26.7	27.6
82	18.7	19.7	20.7	21.7	22.7	23.6	24.6	25.6	26.6	27.6
83	18.7	19.7	20.6	21.6	22.6	23.6	24.6	25.5	26.5	27.5
84	18.6	19.6	20.6	21.6	22.6	23.5	24.5	25.5	26.5	27.4
85	18.6	19.5	20.5	21.5	22.5	23.5	24.5	25.4	26.4	27.4
86	18.5	19.5	20.5	21.4	22.4	23.4	24.4	25.4	26.3	27.3
87	18.4	19.4	20.4	21.4	22.4	23.3	24.3	25.3	26.3	27.3
88	18.4	19.4	20.4	21.3	22.3	23.3	24.3	25.2	26.2	27.2
89	18.3	19.3	20.3	21.3	22.2	23.2	24.2	25.2	26.2	27.1
90	18.3	19.3	20.3	21.2	22.2	23.2	24.2	25.1	26.1	27.1
91	18.2	19.2	20.2	21.2	22.1	23.1	24.1	25.1	26.0	27.0
92	18.2	19.2	20.1	21.1	22.1	23.0	24.0	25.0	26.0	26.9
93	18.1	19.1	20.1	21.0	22.0	23.0	24.0	24.9	25.9	26.9
94	18.1	19.0	20.0	21.0	22.0	22.9	23.9	24.9	25.9	26.8

TABLE XLIX.—REDUCTION OF OBSERVED DEGREES A.P.I. TO DEGREES A.P.I. AT 60°F.—(Continued)

Observed temp., °F.	Observed degrees A.P.I.									
	20	21	22	23	24	25	26	27	28	29
Corresponding degrees A.P.I. at 60°F.										
95	18.0	19.0	20.0	20.9	21.9	22.9	23.8	24.8	25.8	26.8
96	18.0	18.9	19.9	20.9	21.9	22.8	23.8	24.8	25.8	26.7
97	17.9	18.9	19.9	20.8	21.8	22.8	23.7	24.7	25.7	26.6
98	17.9	18.8	19.8	20.8	21.7	22.7	23.7	24.6	25.6	26.6
99	17.8	18.8	19.8	20.7	21.7	22.7	23.6	24.6	25.6	26.5
100	17.8	18.7	19.7	20.7	21.6	22.6	23.6	24.5	25.5	26.5
101	17.7	18.7	19.6	20.6	21.6	22.5	23.5	24.5	25.4	26.4
102	17.6	18.6	19.6	20.5	21.5	22.5	23.4	24.4	25.4	26.3
103	17.6	18.6	19.5	20.5	21.5	22.4	23.4	24.4	25.3	26.3
104	17.5	18.5	19.5	20.4	21.4	22.3	23.3	24.3	25.2	26.2
105	17.5	18.5	19.4	20.4	21.3	22.3	23.3	24.2	25.2	26.1
106	17.4	18.4	19.4	20.3	21.3	22.2	23.2	24.2	25.1	26.1
107	17.4	18.4	19.3	20.3	21.2	22.2	23.2	24.1	25.1	26.0
108	17.3	18.3	19.3	20.2	21.2	22.1	23.1	24.0	25.0	25.9
109	17.3	18.3	19.2	20.2	21.1	22.1	23.0	24.0	24.9	25.9
110	17.2	18.2	19.2	20.1	21.1	22.0	23.0	23.9	24.9	25.8
111	17.2	18.1	19.1	20.1	21.0	22.0	22.9	23.9	24.8	25.8
112	17.1	18.1	19.0	20.0	20.9	21.9	22.9	23.8	24.8	25.7
113	17.1	18.0	19.0	19.9	20.9	21.8	22.8	23.7	24.7	25.6
114	17.0	18.0	18.9	19.9	20.8	21.8	22.8	23.7	24.7	25.6
115	17.0	17.9	18.9	19.8	20.8	21.7	22.7	23.6	24.6	25.5
116	16.9	17.9	18.8	19.8	20.7	21.7	22.6	23.6	24.5	25.5
117	16.9	17.8	18.8	19.7	20.7	21.6	22.6	23.5	24.5	25.4
118	16.8	17.8	18.7	19.7	20.6	21.6	22.5	23.5	24.4	25.3
119	16.8	17.7	18.7	19.6	20.6	21.5	22.5	23.4	24.4	25.3

TABLE L.—SPECIFIC GRAVITIES AND POUNDS PER GALLON CORRESPONDING TO DEGREES A.P.I.

Degrees A.P.I.	Specific gravity at 60°/60°F.	Lb. per gal.	Degrees A.P.I.	Specific gravity at 60°/60°F.	Lb. per gal.
4	1.044	8.694	29	0.8816	7.341
4.5	1.040	8.661	29.5	0.8789	7.318
5	1.037	8.636	30	0.8762	7.296
5.5	1.033	8.603	30.5	0.8735	7.273
6	1.029	8.570	31	0.8708	7.251
6.5	1.025	8.536	31.5	0.8681	7.228
7	1.022	8.512	32	0.8654	7.206
7.5	1.018	8.478	32.5	0.8628	7.184
8	1.014	8.444	33	0.8602	7.163
8.5	1.010	8.411	33.5	0.8576	7.141
9	1.007	8.386	34	0.8550	7.119
9.5	1.003	8.353	34.5	0.8524	7.098
10	1.000	8.328	35	0.8498	7.076
10.5	0.9965	8.299	35.5	0.8473	7.055
11	0.9930	8.270	36	0.8448	7.034
11.5	0.9895	8.241	36.5	0.8423	7.013
12	0.9861	8.212	37	0.8398	6.993
12.5	0.9826	8.183	37.5	0.8373	6.972
13	0.9792	8.155	38	0.8348	6.951
13.5	0.9759	8.127	38.5	0.8324	6.930
14	0.9725	8.099	39	0.8299	6.910
14.5	0.9692	8.071	39.5	0.8275	6.890
15	0.9659	8.044	40	0.8251	6.870
15.5	0.9626	8.016	40.5	0.8227	6.850
16	0.9593	7.989	41	0.8203	6.830
16.5	0.9561	7.962	41.5	0.8178	6.810
17	0.9529	7.935	42	0.8155	6.790
17.5	0.9497	7.909	42.5	0.8132	6.771
18	0.9465	7.882	43	0.8109	6.752
18.5	0.9433	7.856	43.5	0.8086	6.732
19	0.9402	7.830	44	0.8063	6.713
19.5	0.9371	7.804	44.5	0.8040	6.694
20	0.9340	7.778	45	0.8017	6.675
20.5	0.9309	7.752	45.5	0.7994	6.656
21	0.9279	7.727	46	0.7972	6.637
21.5	0.9248	7.701	46.5	0.7949	6.618
22	0.9218	7.676	47	0.7927	6.600
22.5	0.9188	7.651	47.5	0.7905	6.582
23	0.9159	7.627	48	0.7883	6.563
23.5	0.9129	7.602	48.5	0.7861	6.545
24	0.9100	7.578	49	0.7839	6.526
24.5	0.9071	7.554	49.5	0.7818	6.509
25	0.9042	7.529	50	0.7796	6.490
25.5	0.9013	7.505	50.5	0.7775	6.473
26	0.8984	7.481	51	0.7753	6.455
26.5	0.8956	7.458	51.5	0.7732	6.437
27	0.8927	7.434	52	0.7711	6.420
27.5	0.8899	7.410	52.5	0.7690	6.402
28	0.8871	7.387	53	0.7669	6.385
28.5	0.8844	7.364	53.5	0.7649	6.368

TABLE L.—SPECIFIC GRAVITIES AND POUNDS PER GALLON CORRESPONDING TO DEGREES A.P.I.—(Continued)

Degrees A.P.I.	Specific gravity at 60°/60°F.	Lb. per gal.	Degrees A.P.I.	Specific gravity at 60°/60°F.	Lb. per gal.
54	0.7628	6.350	77.5	0.6770	5.635
54.5	0.7608	6.334	78	0.6754	5.622
55	0.7587	6.316	78.5	0.6738	5.608
55.5	0.7567	6.300	79	0.6722	5.595
56	0.7547	6.283	79.5	0.6706	5.582
56.5	0.7527	6.266	80	0.6690	5.568
57	0.7507	6.249	80.5	0.6675	5.558
57.5	0.7487	6.233	81	0.6659	5.542
58	0.7467	6.216	81.5	0.6643	5.529
58.5	0.7447	6.199	82	0.6628	5.516
59	0.7428	6.184	82.5	0.6612	5.503
59.5	0.7408	6.167	83	0.6597	5.491
60	0.7389	6.151	83.5	0.6581	5.477
60.5	0.7370	6.135	84	0.6566	5.465
61	0.7351	6.118	84.5	0.6551	5.453
61.5	0.7332	6.103	85	0.6536	5.440
62	0.7313	6.087	85.5	0.6521	5.427
62.5	0.7294	6.072	86	0.6506	5.415
63	0.7275	6.056	86.5	0.6491	5.402
63.5	0.7256	6.040	87	0.6476	5.390
64	0.7238	6.025	87.5	0.6461	5.377
64.5	0.7219	6.010	88	0.6446	5.365
65	0.7201	5.994	88.5	0.6432	5.353
65.5	0.7183	5.979	89	0.6417	5.341
66	0.7165	5.964	89.5	0.6403	5.329
66.5	0.7146	5.949	90	0.6388	5.316
67	0.7128	5.934	90.5	0.6374	5.305
67.5	0.7111	5.919	91	0.6360	5.293
68	0.7093	5.904	91.5	0.6345	5.281
68.5	0.7075	5.889	92	0.6331	5.269
69	0.7057	5.874	92.5	0.6317	5.257
69.5	0.7040	5.860	93	0.6303	5.245
70	0.7022	5.845	93.5	0.6289	5.234
70.5	0.7005	5.831	94	0.6275	5.222
71	0.6988	5.817	94.5	0.6261	5.211
71.5	0.6970	5.802	95	0.6247	5.199
72	0.6952	5.788	95.5	0.6233	5.187
72.5	0.6936	5.773	96	0.6220	5.176
73	0.6919	5.759	96.5	0.6206	5.164
73.5	0.6902	5.745	97	0.6193	5.154
74	0.6886	5.731	97.5	0.6179	5.142
74.5	0.6869	5.718	98	0.6166	5.131
75	0.6852	5.703	98.5	0.6152	5.120
75.5	0.6836	5.690	99	0.6139	5.109
76	0.6819	5.676	99.5	0.6126	5.098
76.5	0.6803	5.662	100	0.6112	5.086
77	0.6787	5.649			

TABLE LI.—DEGREES A.P.I.—CORRESPONDING TO SPECIFIC GRAVITIES AT
 60° F. (15.56°)

(Calculated from the formula degrees A.P.I.

141.5
specific gravity $60^{\circ}/60^{\circ}$ F. ~~~~)

Sp. gr. $60^{\circ}/60^{\circ}$ F.	0	1	2	3	4	5	6	7	8	9
0.60	104.33	103.94	103.55	103.16	102.77	102.38	102.00	101.61	101.23	100.85
0.61	100.47	100.09	99.71	99.33	98.96	98.58	98.21	97.84	97.46	97.09
0.62	96.73	96.36	95.99	95.63	95.26	94.00	94.54	94.18	93.82	93.46
0.63	93.10	92.75	92.39	92.04	91.69	91.33	90.98	90.63	90.29	89.94
0.64	89.59	89.25	88.90	88.56	88.22	87.88	87.54	87.20	86.86	86.53
0.65	86.19	85.86	85.52	85.19	84.86	84.53	84.20	83.87	83.55	83.22
0.66	82.89	82.57	82.25	81.92	81.60	81.28	80.96	80.64	80.33	80.01
0.67	79.69	79.38	79.07	78.75	78.44	78.13	77.82	77.51	77.20	76.89
0.68	76.59	76.28	75.98	75.67	75.37	75.07	74.77	74.47	74.17	73.87
0.69	73.57	73.28	72.98	72.68	72.39	72.10	71.80	71.51	71.22	70.93
0.70	70.64	70.35	70.07	69.78	69.49	69.21	68.92	68.64	68.36	68.08
0.71	67.80	67.52	67.24	66.96	66.68	66.40	66.13	65.85	65.58	65.30
0.72	65.03	64.76	64.48	64.21	63.94	63.67	63.40	63.14	62.87	62.60
0.73	62.34	62.07	61.81	61.54	61.28	61.02	60.76	60.49	60.23	59.97
0.74	59.72	59.46	59.20	58.94	58.69	58.43	58.18	57.92	57.67	57.42
0.75	57.17	56.92	56.66	56.41	56.17	55.92	55.67	55.42	55.18	54.93
0.76	54.68	54.44	54.20	53.95	53.71	53.47	53.23	52.98	52.74	52.51
0.77	52.27	52.03	51.79	51.55	51.32	51.08	50.85	50.61	50.38	50.14
0.78	49.91	49.68	49.45	49.22	48.98	48.75	48.53	48.30	48.07	47.84
0.79	47.61	47.39	47.16	46.94	46.71	46.49	46.26	46.04	45.82	45.60
0.80	45.38	45.15	44.93	44.71	44.49	44.28	44.06	43.84	43.62	43.41
0.81	43.19	42.98	42.76	42.55	42.33	42.12	41.91	41.69	41.48	41.27
0.82	41.06	40.85	40.64	40.43	40.22	40.02	39.81	39.60	39.39	39.19
0.83	38.98	38.78	38.57	38.37	38.16	37.96	37.76	37.56	37.35	37.15
0.84	36.95	36.75	36.55	36.35	36.15	35.96	35.76	35.56	35.36	35.17
0.85	34.97	34.77	34.58	34.39	34.19	34.00	33.80	33.61	33.42	33.23
0.86	34.03	32.84	32.65	32.46	32.27	32.08	31.89	31.71	31.52	31.33
0.87	31.14	30.96	30.77	30.58	30.40	30.21	30.03	29.85	29.66	29.48
0.88	29.30	29.11	28.93	28.75	28.57	28.39	28.21	28.03	27.85	27.67
0.89	27.49	27.31	27.13	26.95	26.78	26.60	26.42	26.25	26.07	25.90
0.90	25.72	25.55	25.37	25.20	25.03	24.85	24.68	24.51	24.34	24.17
0.91	23.99	23.82	23.65	23.48	23.31	23.14	22.98	22.81	22.64	22.47
0.92	22.30	22.41	21.97	21.80	21.64	21.47	21.31	21.14	20.98	20.81
0.93	20.65	20.49	20.32	20.16	20.00	19.84	19.68	19.51	19.35	19.19
0.94	19.03	18.87	18.71	18.55	18.39	18.24	18.08	17.92	17.76	17.60
0.95	17.45	17.29	17.13	16.98	16.82	16.67	16.51	16.36	16.20	16.06
0.96	15.90	15.74	15.59	15.44	15.28	15.13	14.98	14.83	14.68	14.53
0.97	14.38	14.23	14.08	13.93	13.78	13.63	13.48	13.33	13.18	13.04
0.98	12.89	12.74	12.59	12.45	12.30	12.15	12.01	11.86	11.72	11.57
0.99	11.43	11.29	11.14	11.00	10.85	10.71	10.57	10.43	10.28	10.14
1.00	10.00	9.86	9.72	9.58	9.43	9.30	9.16	9.02	8.88	8.74

TABLE LII.—DEGREES A.P.I. (MODULUS 141.5) CORRESPONDING TO DEGREES BAUMÉ (MODULUS 140), PETROLEUM OILS¹

Degrees Baumé (modulus 140)	Tenths of degrees Baumé									
	0	1	2	3	4	5	6	7	8	9
10	10.00	10.10	10.20	10.30	10.40	10.51	10.61	10.71	10.81	10.91
11	11.01	11.11	11.21	11.31	11.42	11.52	11.62	11.72	11.82	11.92
12	12.02	12.12	12.22	12.32	12.43	12.53	12.63	12.73	12.83	12.93
13	13.03	13.13	13.23	13.34	13.44	13.54	13.64	13.74	13.84	13.94
14	14.04	14.14	14.24	14.35	14.45	14.55	14.65	14.75	14.85	14.95
15	15.05	15.15	15.26	15.36	15.46	15.56	15.66	15.76	15.86	15.96
16	16.06	16.17	16.27	16.37	16.47	16.57	16.67	16.77	16.87	16.97
17	17.08	17.18	17.28	17.38	17.48	17.58	17.68	17.78	17.88	17.98
18	18.09	18.19	18.29	18.39	18.49	18.59	18.69	18.79	18.89	19.00
19	19.10	19.20	19.30	19.40	19.50	19.60	19.70	19.80	19.90	20.01
20	20.11	20.21	20.31	20.41	20.51	20.61	20.71	20.81	20.92	21.02
21	21.12	21.22	21.32	21.42	21.52	21.52	21.72	21.83	21.93	22.03
22	22.13	22.23	22.33	22.43	22.53	22.63	22.74	22.84	22.94	23.04
23	23.14	23.24	23.34	23.44	23.54	23.64	23.75	23.85	23.95	24.05
24	24.15	24.25	24.35	24.45	24.55	24.66	24.76	24.86	24.96	25.06
25	25.16	25.26	25.36	25.46	25.56	25.67	25.77	25.87	25.97	26.07
26	26.17	26.27	26.37	26.47	26.58	26.68	26.78	26.88	26.98	27.08
27	27.18	27.28	27.38	27.49	27.59	27.69	27.79	27.89	27.99	28.09
28	28.19	28.29	28.40	28.50	28.60	28.70	28.80	28.90	29.00	29.10
29	29.20	29.30	29.41	29.51	29.61	29.71	29.81	29.91	30.01	30.11
30	30.21	30.32	30.42	30.52	30.62	30.72	30.82	30.92	31.02	31.12
31	31.22	31.33	31.43	31.53	31.63	31.73	31.83	31.93	32.03	32.13
32	32.24	32.34	32.44	32.54	32.64	32.84	32.84	32.94	33.04	33.15
33	33.25	33.35	33.45	33.55	33.65	33.75	33.85	33.95	34.06	34.16
34	34.26	34.36	34.46	34.56	34.66	34.76	34.86	34.96	35.07	35.17
35	35.27	35.37	35.47	35.57	35.67	35.77	35.87	35.98	36.08	36.18
36	36.28	36.38	36.48	36.58	36.68	36.78	36.88	36.99	37.09	37.19
37	37.29	37.39	37.49	37.59	37.69	37.79	37.90	38.00	38.10	38.20
38	38.30	38.40	38.50	38.60	38.70	38.81	38.91	39.01	39.11	39.21
39	39.31	39.41	39.51	39.61	39.72	39.82	39.92	40.02	40.12	40.22
40	40.32	40.42	40.52	40.62	40.73	40.83	40.93	41.03	41.13	41.23
41	41.33	41.43	41.53	41.64	41.74	41.84	41.94	42.04	42.14	42.24
42	42.34	42.44	42.54	42.65	42.75	42.85	42.95	43.05	43.15	43.24
43	43.35	43.45	43.56	43.66	43.76	43.86	43.96	44.06	44.16	44.26
44	44.36	44.47	44.57	44.67	44.77	44.87	44.97	45.07	45.17	45.27
45	45.38	45.48	45.58	45.68	45.78	45.88	45.98	46.08	46.18	46.28
46	46.39	46.49	46.59	46.69	46.79	46.89	46.99	47.09	47.19	47.30
47	47.40	47.50	47.60	47.70	47.80	47.90	48.00	48.10	48.20	48.31
48	48.41	48.51	48.61	48.71	48.81	48.91	49.01	49.11	49.22	49.32
49	49.42	49.52	49.62	49.72	49.82	49.92	50.02	50.13	50.23	50.33

¹ Extract from Table 6, *Bur. Standards, Circ. 154.*

TABLE LII.—DEGREES A.P.I. (MODULUS 141.5) CORRESPONDING TO DEGREES BAUMÉ (MODULUS 140), PETROLEUM OILS.¹—(Continued)

Degrees Baumé (modulus 140)	Tenths of degrees Baumé									
	0	1	2	3	4	5	6	7	8	9
50	50.43	50.53	50.63	50.73	50.83	50.93	51.04	51.14	51.24	51.34
51	51.44	51.54	51.64	51.74	51.84	51.94	52.05	52.15	52.25	52.35
52	52.45	52.55	52.65	52.75	52.85	52.96	53.06	53.16	53.26	53.36
53	53.46	53.56	53.66	53.76	53.86	53.97	54.07	54.17	54.27	54.37
54	54.47	54.57	54.67	54.77	54.88	54.98	55.08	55.18	55.28	55.38
55	55.48	55.58	55.68	55.79	55.89	55.99	56.09	56.19	56.29	56.39
56	56.49	56.59	56.70	56.80	56.90	57.00	57.10	57.20	57.30	57.40
57	57.50	57.60	57.71	57.81	57.91	58.01	58.11	58.21	58.31	58.41
58	58.51	58.62	58.72	58.82	58.92	59.02	59.12	59.22	59.32	59.42
59	59.52	59.63	59.73	59.83	59.93	60.03	60.13	60.23	60.33	60.43
60	60.54	60.64	60.74	60.84	60.94	61.04	61.14	61.24	61.34	61.45
61	61.55	61.65	61.75	61.85	61.95	62.05	62.15	62.25	62.36	62.46
62	62.56	62.66	62.76	62.86	62.96	63.06	63.16	63.26	63.37	63.47
63	63.57	63.67	63.77	63.87	63.97	64.07	64.17	64.28	64.38	64.48
64	64.58	64.68	64.78	64.88	64.98	65.08	65.18	65.29	65.39	65.49
65	65.59	65.69	65.79	65.89	65.99	66.09	66.20	66.30	66.40	66.50
66	66.60	66.70	66.80	66.90	67.00	67.11	67.21	67.31	67.41	67.51
67	67.61	67.71	67.81	67.91	68.02	68.12	68.22	68.32	68.42	68.52
68	68.62	68.72	68.82	68.92	69.03	69.13	69.23	69.33	69.43	69.53
69	69.63	69.73	69.83	69.94	70.04	70.14	70.24	70.34	70.44	70.54
70	70.64	70.74	70.84	70.95	71.05	71.15	71.25	71.35	71.45	71.55
71	71.65	71.75	71.86	71.96	72.06	72.16	72.26	72.36	72.46	72.56
72	72.66	72.77	72.87	72.97	73.07	73.17	73.27	73.37	73.47	73.57
73	73.68	73.78	73.88	73.98	74.08	74.18	74.28	74.38	74.48	74.58
74	74.69	74.79	74.89	74.99	75.09	75.19	75.29	75.39	75.49	75.60

¹ Extract from Table 6, *Bur. Standards, Circ. 154.*

OIL BURNERS

TABLE LIII.—VOLUME AT 60°F. OCCUPIED BY UNIT VOLUME AT INDICATED TEMPERATURE FOR PETROLEUM OILS

Observed temp., °F.	Degrees A.P.I. at 60°F.									
	10	11	12	13	14	15	16	17	18	19
	Volume at 60°F. occupied by unit volume at indicated temp.									
30	1.0111	1.0112	1.0113	1.0113	1.0114	1.0115	1.0116	1.0117	1.0117	1.0118
31	1.0107	1.0108	1.0109	1.0109	1.0110	1.0111	1.0112	1.0113	1.0113	1.0114
32	1.0103	1.0104	1.0105	1.0105	1.0106	1.0107	1.0108	1.0109	1.0109	1.0110
33	1.0100	1.0101	1.0101	1.0102	1.0102	1.0103	1.0104	1.0105	1.0105	1.0106
34	1.0096	1.0097	1.0097	1.0098	1.0098	1.0099	1.0100	1.0101	1.0101	1.0102
35	1.0092	1.0093	1.0093	1.0094	1.0095	1.0095	1.0096	1.0097	1.0097	1.0098
36	1.0088	1.0089	1.0089	1.0090	1.0090	1.0091	1.0092	1.0093	1.0093	1.0094
37	1.0084	1.0085	1.0085	1.0086	1.0086	1.0087	1.0088	1.0089	1.0089	1.0090
38	1.0081	1.0082	1.0082	1.0083	1.0083	1.0084	1.0085	1.0085	1.0085	1.0086
39	1.0077	1.0078	1.0078	1.0079	1.0079	1.0080	1.0081	1.0081	1.0082	1.0082
40	1.0073	1.0074	1.0074	1.0075	1.0075	1.0076	1.0077	1.0077	1.0078	1.0078
41	1.0069	1.0070	1.0070	1.0071	1.0071	1.0072	1.0073	1.0073	1.0074	1.0074
42	1.0066	1.0066	1.0067	1.0067	1.0068	1.0068	1.0069	1.0069	1.0070	1.0070
43	1.0062	1.0063	1.0063	1.0064	1.0064	1.0065	1.0065	1.0066	1.0066	1.0067
44	1.0059	1.0059	1.0060	1.0060	1.0061	1.0061	1.0061	1.0062	1.0062	1.0063
45	1.0055	1.0055	1.0056	1.0056	1.0057	1.0057	1.0057	1.0058	1.0058	1.0059
46	1.0051	1.0051	1.0052	1.0052	1.0053	1.0053	1.0053	1.0054	1.0054	1.0055
47	1.0048	1.0048	1.0048	1.0049	1.0049	1.0049	1.0049	1.0050	1.0050	1.0051
48	1.0044	1.0044	1.0045	1.0045	1.0046	1.0046	1.0046	1.0046	1.0047	1.0047
49	1.0041	1.0041	1.0042	1.0042	1.0042	1.0042	1.0042	1.0043	1.0043	1.0043
50	1.0037	1.0037	1.0037	1.0038	1.0038	1.0038	1.0038	1.0038	1.0039	1.0039
51	1.0033	1.0033	1.0033	1.0034	1.0034	1.0034	1.0034	1.0034	1.0035	1.0035
52	1.0029	1.0029	1.0029	1.0030	1.0030	1.0030	1.0030	1.0030	1.0031	1.0031
53	1.0026	1.0026	1.0026	1.0027	1.0027	1.0027	1.0027	1.0027	1.0028	1.0028
54	1.0022	1.0022	1.0022	1.0023	1.0023	1.0023	1.0023	1.0023	1.0024	1.0024
55	1.0018	1.0018	1.0019	1.0019	1.0019	1.0019	1.0019	1.0020	1.0020	1.0020
56	1.0014	1.0014	1.0014	1.0015	1.0015	1.0015	1.0015	1.0015	1.0016	1.0016
57	1.0011	1.0011	1.0011	1.0011	1.0011	1.0011	1.0011	1.0011	1.0012	1.0012
58	1.0007	1.0007	1.0007	1.0008	1.0008	1.0008	1.0008	1.0008	1.0008	1.0008
59	1.0004	1.0004	1.0004	1.0004	1.0004	1.0004	1.0004	1.0004	1.0004	1.0004
60	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
61	0.9996	0.9996	0.9996	0.9996	0.9996	0.9996	0.9996	0.9996	0.9996	0.9996
62	0.9993	0.9993	0.9993	0.9992	0.9992	0.9992	0.9992	0.9992	0.9992	0.9992
63	0.9989	0.9989	0.9989	0.9989	0.9989	0.9989	0.9989	0.9989	0.9988	0.9988
64	0.9986	0.9986	0.9986	0.9985	0.9985	0.9985	0.9985	0.9985	0.9984	0.9984
65	0.9982	0.9982	0.9982	0.9981	0.9981	0.9981	0.9981	0.9981	0.9980	0.9980
66	0.9978	0.9978	0.9978	0.9977	0.9977	0.9977	0.9977	0.9977	0.9976	0.9976
67	0.9975	0.9975	0.9974	0.9974	0.9973	0.9973	0.9973	0.9973	0.9972	0.9972
68	0.9971	0.9971	0.9971	0.9970	0.9970	0.9970	0.9970	0.9970	0.9969	0.9969
69	0.9968	0.9967	0.9967	0.9966	0.9966	0.9966	0.9966	0.9966	0.9965	0.9965
70	0.9964	0.9964	0.9963	0.9963	0.9962	0.9962	0.9962	0.9962	0.9961	0.9961
71	0.9960	0.9960	0.9959	0.9959	0.9958	0.9958	0.9958	0.9958	0.9957	0.9957
72	0.9957	0.9956	0.9956	0.9955	0.9955	0.9954	0.9954	0.9954	0.9953	0.9953
73	0.9953	0.9953	0.9952	0.9952	0.9951	0.9951	0.9951	0.9950	0.9950	0.9949
74	0.9950	0.9949	0.9949	0.9948	0.9948	0.9947	0.9947	0.9946	0.9946	0.9945
75	0.9946	0.9945	0.9945	0.9944	0.9944	0.9943	0.9943	0.9942	0.9942	0.9941
76	0.9942	0.9941	0.9941	0.9940	0.9940	0.9939	0.9939	0.9938	0.9938	0.9937
77	0.9939	0.9938	0.9938	0.9937	0.9937	0.9936	0.9935	0.9935	0.9934	0.9934
78	0.9935	0.9934	0.9934	0.9933	0.9933	0.9932	0.9932	0.9931	0.9931	0.9930
79	0.9932	0.9931	0.9931	0.9930	0.9930	0.9929	0.9928	0.9928	0.9927	0.9927
80	0.9928	0.9927	0.9927	0.9926	0.9926	0.9925	0.9924	0.9924	0.9923	0.9923
81	0.9924	0.9923	0.9923	0.9922	0.9922	0.9921	0.9920	0.9920	0.9919	0.9919
82	0.9921	0.9920	0.9919	0.9919	0.9918	0.9917	0.9916	0.9916	0.9915	0.9915
83	0.9917	0.9916	0.9916	0.9915	0.9915	0.9914	0.9913	0.9912	0.9912	0.9911
84	0.9914	0.9913	0.9912	0.9912	0.9911	0.9910	0.9909	0.9908	0.9908	0.9907

TABLE LIII.—VOLUME AT 60°F. OCCUPIED BY UNIT VOLUME AT INDICATED TEMPERATURE FOR PETROLEUM OILS.—(Continued)

Observed temp., °F.	Degrees A.P.I. at 60°F.									
	10	11	12	13	14	15	16	17	18	19
Volume at 60°F. occupied by unit volume at indicated temp.										
85	0.9910	0.9909	0.9908	0.9908	0.9907	0.9906	0.9905	0.9904	0.9904	0.9903
86	0.9906	0.9905	0.9904	0.9904	0.9903	0.9902	0.9901	0.9900	0.9900	0.9899
87	0.9903	0.9902	0.9901	0.9900	0.9899	0.9898	0.9897	0.9896	0.9896	0.9895
88	0.9899	0.9898	0.9897	0.9897	0.9896	0.9895	0.9894	0.9893	0.9893	0.9892
89	0.9896	0.9895	0.9894	0.9893	0.9892	0.9891	0.9890	0.9889	0.9889	0.9888
90	0.9892	0.9891	0.9890	0.9889	0.9888	0.9887	0.9886	0.9885	0.9885	0.9884
91	0.9888	0.9887	0.9886	0.9885	0.9884	0.9883	0.9882	0.9881	0.9881	0.9880
92	0.9885	0.9884	0.9883	0.9881	0.9880	0.9879	0.9878	0.9877	0.9877	0.9876
93	0.9881	0.9880	0.9879	0.9878	0.9877	0.9876	0.9875	0.9874	0.9873	0.9872
94	0.9878	0.9877	0.9876	0.9874	0.9873	0.9872	0.9871	0.9870	0.9869	0.9868
95	0.9874	0.9873	0.9872	0.9870	0.9869	0.9868	0.9867	0.9866	0.9865	0.9864
96	0.9870	0.9869	0.9868	0.9866	0.9865	0.9864	0.9863	0.9862	0.9861	0.9860
97	0.9867	0.9866	0.9865	0.9863	0.9862	0.9861	0.9860	0.9859	0.9857	0.9856
98	0.9863	0.9862	0.9861	0.9859	0.9858	0.9857	0.9856	0.9855	0.9854	0.9853
99	0.9860	0.9859	0.9858	0.9856	0.9855	0.9854	0.9853	0.9852	0.9850	0.9849
100	0.9856	0.9855	0.9854	0.9852	0.9851	0.9850	0.9849	0.9848	0.9846	0.9845
101	0.9852	0.9851	0.9850	0.9848	0.9847	0.9846	0.9845	0.9844	0.9842	0.9841
102	0.9849	0.9848	0.9846	0.9845	0.9843	0.9842	0.9841	0.9840	0.9838	0.9837
103	0.9845	0.9844	0.9843	0.9841	0.9840	0.9839	0.9838	0.9837	0.9835	0.9834
104	0.9842	0.9841	0.9839	0.9838	0.9836	0.9835	0.9834	0.9833	0.9831	0.9830
105	0.9838	0.9837	0.9835	0.9834	0.9832	0.9831	0.9830	0.9829	0.9827	0.9826
106	0.9834	0.9833	0.9831	0.9830	0.9828	0.9827	0.9826	0.9825	0.9823	0.9822
107	0.9831	0.9830	0.9828	0.9827	0.9825	0.9824	0.9823	0.9821	0.9820	0.9818
108	0.9827	0.9826	0.9824	0.9823	0.9821	0.9820	0.9819	0.9818	0.9816	0.9815
109	0.9824	0.9823	0.9821	0.9820	0.9818	0.9817	0.9816	0.9814	0.9813	0.9811
110	0.9820	0.9819	0.9817	0.9816	0.9814	0.9813	0.9812	0.9810	0.9809	0.9807
111	0.9816	0.9815	0.9813	0.9812	0.9810	0.9809	0.9808	0.9806	0.9805	0.9803
112	0.9813	0.9811	0.9810	0.9808	0.9807	0.9805	0.9804	0.9802	0.9801	0.9799
113	0.9809	0.9808	0.9806	0.9805	0.9803	0.9802	0.9800	0.9799	0.9797	0.9796
114	0.9806	0.9804	0.9803	0.9801	0.9800	0.9798	0.9796	0.9795	0.9793	0.9792
115	0.9802	0.9800	0.9799	0.9797	0.9796	0.9794	0.9792	0.9791	0.9789	0.9788
116	0.9799	0.9797	0.9795	0.9794	0.9792	0.9790	0.9788	0.9787	0.9785	0.9784
117	0.9795	0.9793	0.9792	0.9790	0.9789	0.9787	0.9785	0.9783	0.9782	0.9780
118	0.9792	0.9790	0.9788	0.9787	0.9785	0.9783	0.9781	0.9780	0.9778	0.9777
119	0.9788	0.9786	0.9785	0.9783	0.9782	0.9780	0.9778	0.9776	0.9775	0.9773
Observed temp., °F.	Degrees A.P.I. at 60°F.									
	20	21	22	23	24	25	26	27	28	29
Volume at 60°F. occupied by unit volume at indicated temp.										
30	1.0119	1.0120	1.0120	1.0121	0.0122	1.0123	1.0124	1.0125	1.0126	1.0127
31	1.0115	1.0116	1.0116	1.0117	1.0118	1.0119	1.0120	1.0121	1.0122	1.0123
32	1.0111	1.0112	1.0112	1.0113	1.0114	1.0115	1.0116	1.0117	1.0118	1.0118
33	1.0107	1.0108	1.0108	1.0109	1.0109	1.0110	1.0111	1.0112	1.0113	1.0114
34	1.0103	1.0104	1.0104	1.0105	1.0105	1.0106	1.0107	1.0108	1.0109	1.0109
35	1.0099	1.0100	1.0100	1.0101	1.0101	1.0102	1.0103	1.0104	1.0105	1.0105
36	1.0095	1.0096	1.0096	1.0097	1.0097	1.0098	1.0099	1.0100	1.0101	1.0101
37	1.0091	1.0092	1.0092	1.0093	1.0093	1.0094	1.0095	1.0096	1.0097	1.0097
38	1.0087	1.0088	1.0088	1.0089	1.0089	1.0090	1.0091	1.0091	1.0092	1.0092
39	1.0083	1.0084	1.0084	1.0085	1.0085	1.0086	1.0087	1.0087	1.0088	1.0088

TABLE LIII.—VOLUME AT 60°F. OCCUPIED BY UNIT VOLUME AT INDICATED TEMPERATURE FOR PETROLEUM OILS.—(Continued)

Degrees A.P.I. at 60°F.

Observed temp., °F.	20	21	22	23	24	25	26	27	28	29
Volume at 60°F. occupied by unit volume at indicated temp.										
40	1.0079	1.0080	1.0080	1.0081	1.0081	1.0082	1.0083	1.0083	1.0084	1.0084
41	1.0075	1.0076	1.0076	1.0077	1.0077	1.0078	1.0079	1.0079	1.0080	1.0080
42	1.0071	1.0072	1.0072	1.0073	1.0073	1.0074	1.0075	1.0075	1.0076	1.0076
43	1.0067	1.0067	1.0068	1.0068	1.0069	1.0069	1.0070	1.0070	1.0071	1.0071
44	1.0063	1.0063	1.0064	1.0064	1.0064	1.0065	1.0066	1.0066	1.0067	1.0067
45	1.0059	1.0059	1.0060	1.0060	1.0061	1.0062	1.0062	1.0063	1.0063	1.0063
46	1.0055	1.0055	1.0056	1.0056	1.0056	1.0057	1.0058	1.0058	1.0059	1.0059
47	1.0051	1.0051	1.0052	1.0052	1.0052	1.0053	1.0054	1.0054	1.0055	1.0055
48	1.0047	1.0047	1.0048	1.0048	1.0048	1.0049	1.0049	1.0050	1.0050	1.0050
49	1.0043	1.0043	1.0044	1.0044	1.0044	1.0045	1.0045	1.0046	1.0046	1.0046
50	1.0039	1.0039	1.0040	1.0040	1.0040	1.0041	1.0041	1.0042	1.0042	1.0042
51	1.0035	1.0035	1.0036	1.0036	1.0036	1.0037	1.0037	1.0038	1.0038	1.0038
52	1.0031	1.0031	1.0032	1.0032	1.0032	1.0033	1.0033	1.0034	1.0034	1.0034
53	1.0028	1.0028	1.0028	1.0028	1.0028	1.0028	1.0028	1.0029	1.0029	1.0029
54	1.0024	1.0024	1.0024	1.0024	1.0024	1.0024	1.0024	1.0025	1.0025	1.0025
55	1.0020	1.0020	1.0020	1.0020	1.0020	1.0020	1.0020	1.0021	1.0021	1.0021
56	1.0016	1.0016	1.0016	1.0016	1.0016	1.0016	1.0016	1.0017	1.0017	1.0017
57	1.0012	1.0012	1.0012	1.0012	1.0012	1.0012	1.0012	1.0013	1.0013	1.0013
58	1.0008	1.0008	1.0008	1.0008	1.0008	1.0008	1.0008	1.0008	1.0008	1.0008
59	1.0004	1.0004	1.0004	1.0004	1.0004	1.0004	1.0004	1.0004	1.0004	1.0004
60	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
61	0.9996	0.9996	0.9996	0.9996	0.9996	0.9996	0.9996	0.9996	0.9996	0.9996
62	0.999'	0.9992	0.9992	0.9992	0.9992	0.9992	0.9992	0.9992	0.9992	0.9992
63	0.9988	0.9988	0.9988	0.9988	0.9988	0.9988	0.9988	0.9987	0.9987	0.9987
64	0.9984	0.9984	0.9984	0.9984	0.9984	0.9984	0.9984	0.9983	0.9983	0.9983
65	0.9980	0.9980	0.9980	0.9980	0.9980	0.9980	0.9980	0.9979	0.9979	0.9979
66	0.9976	0.9976	0.9976	0.9976	0.9976	0.9976	0.9976	0.9975	0.9975	0.9975
67	0.9972	0.9972	0.9972	0.9972	0.9972	0.9972	0.9972	0.9971	0.9971	0.9971
68	0.9969	0.9969	0.9968	0.9968	0.9968	0.9967	0.9967	0.9966	0.9966	0.9966
69	0.9965	0.9965	0.9964	0.9964	0.9964	0.9963	0.9963	0.9962	0.9962	0.9962
70	0.9961	0.9961	0.9960	0.9960	0.9960	0.9959	0.9959	0.9958	0.9958	0.9958
71	0.9957	0.9957	0.9956	0.9956	0.9956	0.9955	0.9955	0.9954	0.9954	0.9954
72	0.9953	0.9953	0.9952	0.9952	0.9952	0.9951	0.9951	0.9950	0.9950	0.9950
73	0.9949	0.9949	0.9948	0.9948	0.9948	0.9947	0.9947	0.9946	0.9946	0.9946
74	0.9945	0.9945	0.9944	0.9944	0.9944	0.9943	0.9943	0.9942	0.9942	0.9942
75	0.9941	0.9941	0.9940	0.9940	0.9940	0.9939	0.9939	0.9938	0.9938	0.9938
76	0.9937	0.9937	0.9936	0.9936	0.9936	0.9935	0.9935	0.9934	0.9934	0.9934
77	0.9933	0.9933	0.9932	0.9932	0.9932	0.9931	0.9931	0.9930	0.9930	0.9929
78	0.9930	0.9929	0.9929	0.9928	0.9928	0.9927	0.9926	0.9926	0.9925	0.9925
79	0.9926	0.9925	0.9925	0.9924	0.9924	0.9923	0.9922	0.9922	0.9921	0.9920
80	0.9922	0.9921	0.9920	0.9920	0.9920	0.9919	0.9918	0.9918	0.9917	0.9916
81	0.9918	0.9917	0.9917	0.9916	0.9916	0.9915	0.9914	0.9914	0.9913	0.9912
82	0.9914	0.9913	0.9913	0.9912	0.9912	0.9911	0.9910	0.9910	0.9909	0.9908
83	0.9910	0.9909	0.9909	0.9908	0.9908	0.9907	0.9906	0.9906	0.9905	0.9904
84	0.9906	0.9905	0.9905	0.9904	0.9904	0.9903	0.9902	0.9901	0.9900	0.9899
85	0.9902	0.9901	0.9900	0.9900	0.9900	0.9899	0.9898	0.9897	0.9896	0.9895
86	0.9898	0.9897	0.9897	0.9896	0.9896	0.9895	0.9894	0.9893	0.9892	0.9891
87	0.9894	0.9893	0.9893	0.9892	0.9892	0.9891	0.9890	0.9889	0.9888	0.9887
88	0.9891	0.9890	0.9889	0.9888	0.9887	0.9886	0.9885	0.9884	0.9883	0.9882
89	0.9887	0.9886	0.9885	0.9884	0.9883	0.9882	0.9881	0.9880	0.9879	0.9878
90	0.9883	0.9882	0.9881	0.9880	0.9879	0.9878	0.9877	0.9876	0.9875	0.9874
91	0.9879	0.9878	0.9877	0.9876	0.9875	0.9874	0.9873	0.9872	0.9871	0.9870
92	0.9875	0.9874	0.9873	0.9872	0.9871	0.9870	0.9869	0.9868	0.9867	0.9866
93	0.9871	0.9870	0.9869	0.9868	0.9867	0.9866	0.9865	0.9864	0.9863	0.9862
94	0.9867	0.9866	0.9865	0.9864	0.9863	0.9862	0.9861	0.9860	0.9859	0.9858

TABLE LIII.—VOLUME AT 60°F. OCCUPIED BY UNIT VOLUME AT INDICATED TEMPERATURE FOR PETROLEUM OILS.—(Continued)

Degrees A.P.I. at 60°F.

Observed temp., °F.	20	21	22	23	24	25	26	28	29
Volume at 60°F. occupied by unit volume at indicated temp.									
95	0.9862	0.9861	0.9860	0.9859	0.9858	0.9857	0.9856	0.9855	0.9854
96	0.9859	0.9858	0.9857	0.9856	0.9855	0.9854	0.9853	0.9852	0.9851
97	0.9855	0.9854	0.9853						
98	0.9852	0.9851	0.9849						
99	0.9848	0.9847	0.9845	0.9844	0.9843	0.9842	0.9841	0.9839	0.9837
100	0.9844	0.9843	0.9841	0.9840	0.9839	0.9838	0.9837	0.9835	0.9834
101	0.9840	0.9839	0.9837	0.9836	0.9835	0.9834	0.9833	0.9831	0.9830
102	0.9836	0.9835	0.9833	0.9832	0.9831	0.9830	0.9829	0.9827	0.9826
103	0.9833	0.9832	0.9830	0.9829	0.9828	0.9826	0.9824	0.9823	0.9821
104	0.9829	0.9828	0.9826	0.9825	0.9824	0.9822	0.9820	0.9819	0.9817
105	0.9825	0.9824	0.9822	0.9821	0.9820	0.9818	0.9816	0.9815	0.9813
106	0.9821	0.9820	0.9818	0.9817	0.9816	0.9814	0.9812	0.9811	0.9809
107	0.9817	0.9816	0.9814	0.9813	0.9812	0.9810	0.9808	0.9807	0.9805
108	0.9814	0.9812	0.9811	0.9809	0.9808	0.9806	0.9804	0.9803	0.9801
109	0.9810	0.9808	0.9807	0.9805	0.9804	0.9802	0.9800	0.9799	0.9797
110	0.9806	0.9804	0.9803	0.9801	0.9800	0.9798	0.9796	0.9795	0.9793
111	0.9802	0.9800	0.9799	0.9797	0.9796	0.9794	0.9792	0.9791	0.9789
112	0.9798	0.9796	0.9795	0.9793	0.9792	0.9790	0.9788	0.9787	0.9785
113	0.9794	0.9792	0.9791	0.9789	0.9788	0.9786	0.9784	0.9782	0.9790
114	0.9790	0.9788	0.9787	0.9785	0.9784	0.9782	0.9780	0.9778	0.9776
115	0.9786	0.9784	0.9783	0.9781	0.9780	0.9778	0.9776	0.9774	0.9772
116	0.9782	0.9780	0.9778	0.9777	0.9776	0.9774	0.9772	0.9770	0.9768
117	0.9778	0.9776	0.9775	0.9773	0.9772	0.9770	0.9768	0.9766	0.9764
118	0.9775	0.9773	0.9771	0.9769	0.9768	0.9766	0.9764	0.9761	0.9759
119	0.9771	0.9769	0.9767	0.9765	0.9764	0.9762	0.9760	0.9757	0.9755

TABLE LIV.—EQUIVALENT OF DEGREES BAUMÉ (AMERICAN STANDARD)
AND SPECIFIC GRAVITY AT 60°F.*

$$\text{Degrees Baumé} = 145 - \frac{145}{\text{sp. gr.}}$$

for liquids heavier than water

Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity
0.0	1.0000	4.0	1.0284	8.0	1.0584	12.0	1.0902
0.1	1.0007	4.1	1.0291	8.1	1.0592	12.1	1.0910
0.2	1.0014	4.2	1.0298	8.2	1.0599	12.2	1.0919
0.3	1.0021	4.3	1.0306	8.3	1.0607	12.3	1.0927
0.4	1.0028	4.4	1.0313	8.4	1.0615	12.4	1.0935
0.5	1.0035	4.5	1.0320	8.5	1.0623	12.5	1.0943
0.6	1.0042	4.6	1.0328	8.6	1.0630	12.6	1.0952
0.7	1.0049	4.7	1.0335	8.7	1.0638	12.7	1.0960
0.8	1.0055	4.8	1.0342	8.8	1.0646	12.8	1.0968
0.9	1.0062	4.9	1.0350	8.9	1.0654	12.9	1.0977
1.0	1.0069	5.0	1.0357	9.0	1.0662	13.0	1.0985
1.1	1.0076	5.1	1.0365	9.1	1.0670	13.1	1.0993
1.2	1.0083	5.2	1.0372	9.2	1.0677	13.2	1.1002
1.3	1.0090	5.3	1.0379	9.3	1.0685	13.3	1.1010
1.4	1.0097	5.4	1.0387	9.4	1.0693	13.4	1.1018
1.5	1.0105	5.5	1.0394	9.5	1.0701	13.5	1.1027
1.6	1.0112	5.6	1.0402	9.6	1.0709	13.6	1.1035
1.7	1.0119	5.7	1.0409	9.7	1.0717	13.7	1.1043
1.8	1.0126	5.8	1.0417	9.8	1.0725	13.8	1.1052
1.9	1.0133	5.9	1.0424	9.9	1.0733	13.9	1.1060
2.0	1.0140	6.0	1.0432	10.0	1.0741	14.0	1.1069
2.1	1.0147	6.1	1.0436	10.1	1.0749	14.1	1.1077
2.2	1.0154	6.2	1.0447	10.2	1.0757	14.2	1.1086
2.3	1.0161	6.3	1.0454	10.3	1.0765	14.3	1.1094
2.4	1.0168	6.4	1.0462	10.4	1.0773	14.4	1.1103
2.5	1.0175	6.5	1.0469	10.5	1.0781	14.5	1.1111
2.6	1.0183	6.6	1.0477	10.6	1.0789	14.6	1.1120
2.7	1.0190	6.7	1.0484	10.7	1.0797	14.7	1.1128
2.8	1.0197	6.8	1.0492	10.8	1.0805	14.8	1.1137
2.9	1.0204	6.9	1.0500	10.9	1.0813	14.9	1.1145
3.0	1.0211	7.0	1.0507	11.0	1.0821	15.0	1.1154
3.1	1.0218	7.1	1.0515	11.1	1.0829	15.1	1.1162
3.2	1.0226	7.2	1.0522	11.2	1.0837	15.2	1.1171
3.3	1.0233	7.3	1.0530	11.3	1.0845	15.3	1.1180
3.4	1.0240	7.4	1.0538	11.4	1.0853	15.4	1.1188
3.5	1.0247	7.5	1.0545	11.5	1.0861	15.5	1.1197
3.6	1.0255	7.6	1.0553	11.6	1.0870	15.6	1.1206
3.7	1.0262	7.7	1.0561	11.7	1.0878	15.7	1.1214
3.8	1.0269	7.8	1.0569	11.8	1.0886	15.8	1.1223
3.9	1.0276	7.9	1.0576	11.9	1.0894	15.9	1.1232

* From Cross, "Handbook of Petroleum, Asphalt and Natural Gas," Kansas City Testing Laboratory.

TABLE LIV.—EQUIVALENT OF DEGREES BAUMÉ (AMERICAN STANDARD)
AND SPECIFIC GRAVITY AT 60°F.*—(Continued)

Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity
16.0	1.1240	20.5	1.1647	25.0	1.2083	29.5	1.2554
16.1	1.1249	20.6	1.1656	25.1	1.2093	29.6	1.2565
16.2	1.1258	20.7	1.1665	25.2	1.2104	29.7	1.2576
16.3	1.1267	20.8	1.1675	25.3	1.2114	29.8	1.2587
16.4	1.1275	20.9	1.1684	25.4	1.2124	29.9	1.2598
16.5	1.1284	21.0	1.1694	25.5	1.2134	30.0	1.2609
16.6	1.1293	21.1	1.1703	25.6	1.2144	30.1	1.2620
16.7	1.1302	21.2	1.1712	25.7	1.2154	30.2	1.2631
16.8	1.1310	21.3	1.1722	25.8	1.2164	30.3	1.2642
16.9	1.1319	21.4	1.1731	25.9	1.2175	30.4	1.2653
17.0	1.1328	21.5	1.1741	26.0	1.2185	30.5	1.2664
17.1	1.1337	21.6	1.1750	26.1	1.2195	30.6	1.2675
17.2	1.1346	21.7	1.1760	26.2	1.2205	30.7	1.2686
17.3	1.1355	21.8	1.1769	26.3	1.2216	30.8	1.2697
17.4	1.1364	21.9	1.1779	26.4	1.2226	30.9	1.2708
17.5	1.1373	22.0	1.1789	26.5	1.2236	31.0	1.2719
17.6	1.1381	22.1	1.1798	26.6	1.2247	31.1	1.2730
17.7	1.1390	22.2	1.1808	26.7	1.2257	31.2	1.2742
17.8	1.1399	22.3	1.1817	26.8	1.2267	31.3	1.2753
17.9	1.1408	22.4	1.1827	26.9	1.2278	31.4	1.2764
18.0	1.1417	22.5	1.1837	27.0	1.2288	31.5	1.2775
18.1	1.1426	22.6	1.1846	27.1	1.2299	31.6	1.2787
18.2	1.1435	22.7	1.1856	27.2	1.2309	31.7	1.2798
18.3	1.1444	22.8	1.1866	27.3	1.2319	31.8	1.2809
18.4	1.1453	22.9	1.1876	27.4	1.2330	31.9	1.2821
18.5	1.1462	23.0	1.1885	27.5	1.2340	32.0	1.2832
18.6	1.1472	23.1	1.1895	27.6	1.2351	32.1	1.2843
18.7	1.1481	23.2	1.1905	27.7	1.2361	32.2	1.2855
18.8	1.1490	23.3	1.1915	27.8	1.2372	32.3	1.2866
18.9	1.1499	23.4	1.1924	27.9	1.2383	32.4	1.2877
19.0	1.1508	23.5	1.1934	28.0	1.2393	32.5	1.2889
19.1	1.1517	23.6	1.1944	28.1	1.2404	32.6	1.2900
19.2	1.1526	23.7	1.1954	28.2	1.2414	32.7	1.2912
19.3	1.1535	23.8	1.1964	28.3	1.2425	32.8	1.2923
19.4	1.1545	23.9	1.1974	28.4	1.2436	32.9	1.2935
19.5	1.1554	24.0	1.1983	28.5	1.2446	33.0	1.2946
19.6	1.1563	24.1	1.1993	28.6	1.2457	33.1	1.2958
19.7	1.1572	24.2	1.2003	28.7	1.2468	33.2	1.2970
19.8	1.1581	24.3	1.2013	28.8	1.2478	33.3	1.2981
19.9	1.1591	24.4	1.2023	28.9	1.2489	33.4	1.2993
20.0	1.1600	24.5	1.2033	29.0	1.2500	33.5	1.3004
20.1	1.1609	24.6	1.2043	29.1	1.2511	33.6	1.3016
20.2	1.1619	24.7	1.2053	29.2	1.2522	33.7	1.3028
20.3	1.1628	24.8	1.2063	29.3	1.2532	33.8	1.3040
20.4	1.1637	24.9	1.2073	29.4	1.2543	33.9	1.3051

TABLE LIV.—EQUIVALENT OF DEGREES BAUMÉ (AMERICAN STANDARD) AND SPECIFIC GRAVITY AT 60°F.*—(Continued)

Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity
34.0	1.3063	38.5	1.3615	43.0	1.4216	47.5	1.4872
34.1	1.3075	38.6	1.3628	43.1	1.4230	47.6	1.4887
34.2	1.3087	38.7	1.3641	43.2	1.4244	47.7	1.4902
34.3	1.3098	38.8	1.3653	43.3	1.4258	47.8	1.4918
34.4	1.3110	38.9	1.3666	43.4	1.4272	47.9	1.4933
34.5	1.3122	39.0	1.3679	43.5	1.4286	48.0	1.4948
34.6	1.3134	39.1	1.3692	43.6	1.4300	48.1	1.4964
34.7	1.3146	39.2	1.3705	43.7	1.4314	48.2	1.4979
34.8	1.3158	39.3	1.3718	43.8	1.4328	48.3	1.4995
34.9	1.3170	39.4	1.3731	43.9	1.4342	48.4	1.5010
35.0	1.3182	39.5	1.3744	44.0	1.4356	48.5	1.5026
35.1	1.3194	39.6	1.3757	44.1	1.4371	48.6	1.5041
35.2	1.3206	39.7	1.3770	44.2	1.4385	48.7	1.5057
35.3	1.3218	39.8	1.3783	44.3	1.4399	48.8	1.5073
35.4	1.3230	39.9	1.3796	44.4	1.4414	48.9	1.5088
35.5	1.3242	40.0	1.3810	44.5	1.4428	49.0	1.5104
35.6	1.3254	40.1	1.3823	44.6	1.4442	49.1	1.5120
35.7	1.3266	40.2	1.3836	44.7	1.4457	49.2	1.5136
35.8	1.3278	40.3	1.3849	44.8	1.4471	49.3	1.5152
35.9	1.3291	40.4	1.3862	44.9	1.4486	49.4	1.5167
36.0	1.3303	40.5	1.3876	45.0	1.4500	49.5	1.5183
36.1	1.3315	40.6	1.3889	45.1	1.4515	49.6	1.5199
36.2	1.3327	40.7	1.3902	45.2	1.4529	49.7	1.5215
36.3	1.3339	40.8	1.3916	45.3	1.4544	49.8	1.5231
36.4	1.3352	40.9	1.3928	45.4	1.4558	49.9	1.5247
36.5	1.3364	41.0	1.3942	45.5	1.4573	50.0	1.5263
36.6	1.3376	41.1	1.3956	45.6	1.4588	50.1	1.5279
36.7	1.3389	41.2	1.3969	45.7	1.4602	50.2	1.5295
36.8	1.3401	41.3	1.3983	45.8	1.4617	50.3	1.5312
36.9	1.3414	41.4	1.3996	45.9	1.4632	50.4	1.5328
37.0	1.3426	41.5	1.4010	46.0	1.4646	50.5	1.5344
37.1	1.3438	41.6	1.4023	46.1	1.4661	50.6	1.5360
37.2	1.3451	41.7	1.4037	46.2	1.4676	50.7	1.5376
37.3	1.3463	41.8	1.4050	46.3	1.4691	50.8	1.5393
37.4	1.3476	41.9	1.4064	46.4	1.4706	50.9	1.5409
37.5	1.3488	42.0	1.4078	46.5	1.4721	51.0	1.5426
37.6	1.3501	42.1	1.4091	46.6	1.4736	51.1	1.5442
37.7	1.3514	42.2	1.4105	46.7	1.4751	51.2	1.5458
37.8	1.3526	42.3	1.4119	46.8	1.4766	51.3	1.5475
37.9	1.3539	42.4	1.4133	46.9	1.4781	51.4	1.5491
38.0	1.3551	42.5	1.4146	47.0	1.4796	51.5	1.5508
38.1	1.3564	42.6	1.4160	47.1	1.4811	51.6	1.5525
38.2	1.3577	42.7	1.4174	47.2	1.4826	51.7	1.5541
38.3	1.3590	42.8	1.4188	47.3	1.4841	51.8	1.5558
38.4	1.3602	42.9	1.4202	47.4	1.4857	51.9	1.5575

TABLE LIV.—EQUIVALENT OF DEGREES BAUMÉ (AMERICAN STANDARD)
AND SPECIFIC GRAVITY AT 60°F.*—(Continued)

Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity
52.0	1.5591	56.5	1.6384	61.0	1.7262	65.5	1.8239
52.1	1.5608	56.6	1.6403	61.1	1.7282	65.6	1.8262
52.2	1.5625	56.7	1.6421	61.2	1.7303	65.7	1.8285
52.3	1.5642	56.8	1.6440	61.3	1.7324	65.8	1.8308
52.4	1.5659	56.9	1.6459	61.4	1.7344	65.9	1.8331
52.5	1.5676	57.0	1.6477	61.5	1.7365	66.0	1.8354
52.6	1.5693	57.1	1.6496	61.6	1.7386	66.1	1.8378
52.7	1.5710	57.2	1.6515	61.7	1.7407	66.2	1.8401
52.8	1.5727	57.3	1.6534	61.8	1.7428	66.3	1.8424
52.9	1.5744	57.4	1.6553	61.9	1.7449	66.4	1.8448
53.0	1.5761	57.5	1.6571	62.0	1.7470	66.5	1.8471
53.1	1.5778	57.6	1.6590	62.1	1.7491	66.6	1.8495
53.2	1.5795	57.7	1.6609	62.2	1.7512	66.7	1.8519
53.3	1.5812	57.8	1.6628	62.3	1.7533	66.8	1.8542
53.4	1.5830	57.9	1.6659	62.4	1.7554	66.9	1.8566
53.5	1.5847	58.0	1.6667	62.5	1.7576	67.0	1.8590
53.6	1.5864	58.1	1.6686	62.6	1.7597	67.1	1.8614
53.7	1.5882	58.2	1.6705	62.7	1.7618	67.2	1.8638
53.8	1.5899	58.3	1.6724	62.8	1.7640	67.3	1.8662
53.9	1.5917	58.4	1.6744	62.9	1.7661	67.4	1.8686
54.0	1.5934	58.5	1.6763	63.0	1.7683	67.5	1.8710
54.1	1.5952	58.6	1.6782	63.1	1.7705	67.6	1.8734
54.2	1.5969	58.7	1.6802	63.2	1.7726	67.7	1.8758
54.3	1.5987	58.8	1.6821	63.3	1.7748	67.8	1.8782
54.4	1.6004	58.9	1.6841	63.4	1.7770	67.9	1.8807
54.5	1.6022	59.0	1.6860	63.5	1.7791	68.0	1.8831
54.6	1.6040	59.1	1.6880	63.6	1.7813	68.1	1.8856
54.7	1.6058	59.2	1.6900	63.7	1.7835	68.2	1.8880
54.8	1.6075	59.3	1.6919	63.8	1.7857	68.3	1.8905
54.9	1.6093	59.4	1.6939	63.9	1.7879	68.4	1.8930
55.0	1.6111	59.5	1.6959	64.0	1.7901	68.5	1.8954
55.1	1.6129	59.6	1.6979	64.1	1.7923	68.6	1.8979
55.2	1.6147	59.7	1.6999	64.2	1.7946	68.7	1.9004
55.3	1.6165	59.8	1.7019	64.3	1.7968	68.8	1.9029
55.4	1.6183	59.9	1.7039	64.4	1.7990	68.9	1.9054
55.5	1.6201	^ ^ ^	1.7059	64.5	1.8012	69.0	1.9079
55.6	1.6219	60.1	1.7079	64.6	1.8035	69.1	1.9104
55.7	1.6237	60.2	1.7099	64.7	1.8057	69.2	1.9129
55.8	1.6256	60.3	1.7119	64.8	1.8080	69.3	1.9155
55.9	1.6274	60.4	1.7139	64.9	1.8102	69.4	1.9180
56.0	1.6292	60.5	1.7160	65.0	1.8125	69.5	1.9205
56.1	1.6310	60.6	1.7180	65.1	1.8148	69.6	1.9231
56.2	1.6329	60.7	1.7200	65.2	1.8170	69.7	1.9256
56.3	1.6347	60.8	1.7221	65.3	1.8193	69.8	1.9282
56.4	1.6366	60.9	1.7241	65.4	1.8216	69.9	1.9308

70.0 1.9333

TABLE LV.—MEAN SPECIFIC HEATS OF VARIOUS SOLIDS AND LIQUIDS
BETWEEN 32 AND 212°F.*

	SOLIDS	Crown.....	0.16	Wood:
Alloys:		Flint.....	0.12	Fir..... 0.65
Bismuth-tin..	0.040-0.045	Gneiss.....	0.18	Oak..... 0.57
Bell metal.....	0.086	Granite.....	0.195	Pine..... 0.67
Brass, yellow....	0.0883	Graphite.....	0.201	LIQUIDS
Brass, red.....	0.090	Gypsum.....	0.259	Acetic acid..... 0.51
Bronze.....	0.104	Hornblende.....	0.195	Alcohol (absolute).... 0.58
Constantan.....	0.098	Humus (soil).....	0.44	Aniline..... 0.49
D'Arcet's metal....	0.050	Ice.....	0.504	Benzol..... 0.40
German silver.....	0.095	India rubber (Para).....		Chloroform..... 0.23
Lipowitz's metal...	0.040		0.27-0.48	Ether..... 0.54
Nickel steel.....	0.109	Kaolin.....	0.224	Fusel oil..... 0.56
Rose's metal.....	0.050	Limestone.....	0.217	Gasoline..... 0.50
Solders (Pb and Sn)		Marble.....	0.210	Glycerine..... 0.58
	0.040-0.045	Oxides:		Hydrochloric acid.... 0.63
Type metal.....	0.0388	Alumina (Al_2O_3)....	0.183	Kerosene..... 0.50
Wood's metal.....	0.040	Cu_2O	0.111	Naphthalene..... 0.31
40Pb + 60Bi.....	0.0317	Lead oxide (PbO)..	0.055	Machine oil..... 0.40
25Pb + 75Bi.....	0.030	Lodestone.....	0.156	Mercury..... 0.033
Asbestos.....	0.20	Magnesia.....	0.222	Olive oil..... 0.40
Ashes.....	0.20	Magnetite (Fe_3O_4)..	0.168	Paraffin oil..... 0.52
Basalt (lava).....	0.20	Silica.....	0.191	Petroleum..... 0.50
Borax.....	0.229	Soda.....	0.231	Sulphuric acid..... 0.836
Brick.....	0.22	Zinc oxide (ZnO)..	0.125	Sea water..... 0.94
Carbon-coke.....	0.203	Paraffin wax.....	0.69	Toluene..... 0.40
Chalk.....	0.215	Porcelain.....	0.22	Turpentine..... 0.42
Charcoal.....	0.20	Quartz.....	0.17-0.28	Molten metals:
Cinders.....	0.18	Quicklime.....	0.217	Bismuth (535-725°F.)
Coal.....	0.314	Salt, rock.....	0.21	
Concrete.....	0.156	Sand.....	0.195	Lead (590-680°F.)
Cork.....	0.485	Sandstone.....	0.22	
Corundum.....	0.198	Serpentine.....	0.25	Sulphur (246-297°F.)
Dolomite.....	0.222	Sulphur.....	0.180	
Ebonite.....	0.33	Talc.....	0.209	Tin (460-660°F.).. 0.058
Glass:		Tufa.....	0.33	
Normal.....	0.199	Vulcanite.....	0.331	

Mean Specific Heat of Iron (c_m) between 32 and $t^{\circ}\text{F}$. (Oberhoffer)

600	800	1000	1200	1400	1600	1800	2000	2250	2500
0.127	0.133	0.139	0.148 ¹	0.167	0.170 ¹	0.169	0.168 ¹	0.167	0.167

* From Marks, "Mechanical Engineers' Handbook," McGraw-Hill Book Company, Inc

TABLE LVI.—PHYSICAL CONSTANTS OF METALS*

Name	Symbol	Atomic weight	Specific gravity at 68° F. †	Specific heat near room temp.	Melting point,	10,000 × coefficient of linear expansion per °F., at 70° F.	Relative electrical conductivity (copper = 100)
Aluminum.....	Al	27.0	2.70	0.207	1220	0.128	64.5
Antimony.....	Sb	121.8	6.71	0.0493	1167	0.0633	4.3
Arsenic.....	As	75.0	5.73	0.0823	1500	0.026	4.8
Barium.....	Ba	137.4	3.5	0.07	1562		
Beryllium.....	Be	9.0	1.84	0.43?	2350	0.068	9.1
Bismuth.....	Bi	209.0	9.80	0.0291	520	0.074	1.5
Boron.....	B	10.8	2.3	0.31?	4200	0.01(?)	9×10^{-13}
Cadmium.....	Cd	112.4	8.65	0.0548	610	0.166	23.
Cæsium.....	Cs	132.8	1.90	0.0523	79	0.54	8.5
Calcium.....	Ca	40.1	1.55	0.152	1490	0.14	37.
Cerium.....	Ce	140.3	6.9	0.0511	1184	2.2
Chromium.....	Cr	52.0	7.1	0.106	2940	0.046	65.
Cobalt.....	Co	59.0	8.9	0.0988	2696	0.068	17.
Copper.....	Cu	63.6	8.94	0.0915	1981	0.0923	100.
Gallium.....	Ga	69.7	5.91	0.079	86	0.100	3.2
Germanium.....	Ge	72.4	5.36	0.074	1756	0.002
Gold.....	Au	197.2	19.30	0.0308	1945	0.0789	70.
Indium.....	In	114.8	7.31	0.0570	311	0.18	19.
Iridium.....	Ir	193.1	22.4	0.0323	4260	0.036	28.
Iron.....	Fe	55.8	7.90	0.101	2795	0.0650	16.9
Lead.....	Pb	207.2	11.35	0.0302	622	0.162	7.71
Lithium.....	Li	6.9	0.53	1.10?	367	0.31	18.
Magnesium.....	Mg	24.3	1.74	0.241	1204	0.142	37.9
Manganese.....	Mn	54.9	7.2	0.107	2300	0.13	34.
Mercury.....	Hg	200.6	13.55	0.0331	-38	1.76
Molybdenum.....	Mo	96.0	10.2	0.065	4750	0.02	35.4
Nickel.....	Ni	58.7	8.90	0.103	2646	0.0712	25.
Osmium.....	Os	190.8	22.48	0.031	4900	0.034	19.
Palladium.....	Pd	106.7	12.1	0.060	2831	0.0656	15.7
Platinum.....	Pt	195.2	21.45	0.0315	3190	0.049	16.1
Potassium.....	K	39.1	0.86	0.177	144	0.46	24.
Rhodium.....	Rh	102.9	12.2	0.058	3550	0.047	33.
Rubidium.....	Rb	85.4	1.53	0.084	101	0.50	13.5
Ruthenium.....	Ru	101.7	12.2	0.061	4440	0.051	17.
Selenium.....	Se	79.2	4.8	0.077	428	0.21	14.0
Silicon.....	Si	28.1	2.4	0.174	2590	0.039	0.002
Silver.....	Ag	107.9	10.49	0.056	1761	0.105	104.
Sodium.....	Na	23.0	0.97	0.29	208	0.39	37.
Strontium.....	Sr	87.6	2.6	1500	7.4
Tantalum.....	Ta	181.5	16.6	0.033	5160	0.04	11.
Tellurium.....	Te	127.5	6.24	0.048	846	0.093	0.03
Thallium.....	Tl	204.4	11.85	0.033	578	0.16	9.34
Thorium.....	Th	232.2	11.7	0.028	3350		9.4
Tin.....	Sn	118.7	7.30	0.053	449	0.11	14.8
Titanium.....	Ti	47.9	4.5	0.143	3300	56.
Tungsten.....	W	184.0	19.3	0.034	6100	0.02	30.8
Uranium.....	U	238.2	18.7	0.028	23360		2.8
Vanadium.....	V	51.0	5.96	0.11	3110		
Zinc.....	Zn	65.4	7.14	0.092	787	0.18	30.
Zirconium.....	Zr	91.	6.4	0.067	3100		1.

* From Marks, "Mechanical Engineers' Handbook," McGraw-Hill Book Company, Inc.

† Water at 4°C. as unity. ‡ Under 36 atm.

TABLE LVII.—WIRE AND SHEET METAL GAUGES*
Diameters and Thicknesses in Inches

Gauge No.	American wire gauge or Brown & Sharpe (for nonferrous sheet and wire)	Steel wire gauge or Washburn & Moen or Roebling (for steel wire)	Birmingham wire gauge (B.W.G.) or Stubs' iron wire (for steel rods or sheets)	Stubs' steel wire gauge	British Imperial standard wire gauge (S.W.G.)	U. S. standard gauge for sheet metal (iron and steel) 1893	Trenton Iron Co.	British standard for iron and steel sheets and hoops, 1914 (B.G.)
0000000	0.4900	0.500	0.500	0.6666
000000	0.4615	0.464	0.469	0.6250
00000	0.4305	0.432	0.438	0.5883
0000	0.460	0.3938	0.454	0.400	0.406	0.400	0.5416
000	0.410	0.3625	0.425	0.372	0.375	0.360	0.5000
00	0.365	0.3310	0.380	0.348	0.344	0.330	0.4452
0	0.325	0.3065	0.340	0.324	0.312	0.305	0.3964
1	0.289	0.2830	0.300	0.227	0.300	0.281	0.285	0.3532
2	0.258	0.2625	0.284	0.219	0.276	0.266	0.265	0.3147
3	0.229	0.2437	0.259	0.212	0.252	0.250	0.245	0.2804
4	0.204	0.2253	0.238	0.207	0.232	0.234	0.225	0.2500
5	0.182	0.2070	0.220	0.204	0.212	0.219	0.205	0.2225
6	0.162	0.1920	0.203	0.201	0.192	0.203	0.190	0.1981
7	0.144	0.1770	0.180	0.199	0.176	0.188	0.175	0.1764
8	0.128	0.1620	0.165	0.197	0.160	0.172	0.160	0.1570
9	0.114	0.1483	0.148	0.194	0.144	0.156	0.145	0.1398
10	0.102	0.1350	0.134	0.191	0.128	0.141	0.130	0.1250
11	0.091	0.1205	0.120	0.188	0.116	0.125	0.1175	0.1113
12	0.081	0.1055	0.109	0.185	0.104	0.109	0.105	0.0991
13	0.072	0.0915	0.095	0.182	0.092	0.094	0.0925	0.0882
14	0.064	0.0800	0.083	0.180	0.080	0.078	0.080	0.0785
15	0.057	0.0720	0.072	0.178	0.072	0.070	0.070	0.0699
16	0.051	0.0625	0.065	0.175	0.064	0.062	0.061	0.0625
17	0.045	0.0540	0.058	0.172	0.056	0.056	0.0525	0.0556
18	0.040	0.0475	0.049	0.168	0.048	0.050	0.045	0.0495
19	0.036	0.0410	0.042	0.164	0.040	0.0438	0.040	0.0440
20	0.032	0.0348	0.035	0.161	0.036	0.0375	0.035	0.0392
21	0.0285	0.0317	0.032	0.157	0.032	0.0344	0.031	0.0349
22	0.0253	0.0286	0.028	0.155	0.028	0.0312	0.028	0.0313
23	0.0226	0.0258	0.025	0.153	0.024	0.0281	0.025	0.0278
24	0.0201	0.0230	0.022	0.151	0.022	0.0250	0.0225	0.0248
25	0.0179	0.0204	0.020	0.148	0.020	0.0219	0.020	0.0220
26	0.0159	0.0181	0.018	0.146	0.018	0.0188	0.018	0.0196
27	0.0142	0.0173	0.016	0.143	0.0164	0.0172	0.017	0.0175
28	0.0126	0.0162	0.014	0.139	0.0148	0.0156	0.016	0.0156
29	0.0113	0.0150	0.013	0.134	0.0136	0.0141	0.015	0.0139
30	0.0100	0.0140	0.012	0.127	0.0124	0.0125	0.014	0.0123
31	0.0089	0.0132	0.010	0.120	0.0116	0.0109	0.013	0.0110
32	0.0080	0.0128	0.009	0.115	0.0108	0.0102	0.012	0.0098
33	0.0071	0.0118	0.008	0.112	0.0100	0.0094	0.011	0.0087
34	0.0063	0.0104	0.007	0.110	0.0092	0.0086	0.010	0.0077
35	0.0056	0.0095	0.005	0.108	0.0084	0.0078	0.0095	0.0069
36	0.0050	0.0090	0.004	0.106	0.0076	0.0070	0.009	0.0061
37	0.0045	0.0085	0.103	0.0068	0.0066	0.0085	0.0054
38	0.0040	0.0080	0.101	0.0060	0.0062	0.008	0.0048
39	0.0035	0.0075	0.099	0.0052	0.0075	0.0043
40	0.0031	0.0070	0.097	0.0048	0.007	0.0039
41	0.0066	0.095	0.0044	0.0034
42	0.0062	0.092	0.0040	0.0031
43	0.0060	0.088	0.0036	0.0027
44	0.0058	0.085	0.0032	0.0024
45	0.0055	0.081	0.0028	0.0022
46	0.0052	0.079	0.0024	0.0019
47	0.0050	0.077	0.0020	0.0017
48	0.0048	0.075	0.0016	0.0015
49	0.0046	0.072	0.0012	0.0014
50	0.0044	0.069	0.0010	0.0012

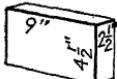
* From Marks, "Mechanical Engineers' Handbook," McGraw-Hill Book Company, Inc.

TABLE LVIII.—FIREBRICK INFORMATION

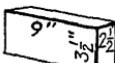
1 sq. ft. of 4½-in. wall requires 7 nine-inch straight brick
1 sq. ft. of 9-in. wall requires 14 nine-inch straight brick
1 sq. ft. of 13½-in. wall requires 21 nine-inch straight brick
1 cu. ft. of firebrick requires 17 brick, and weighs 130 to 140 lb.
1000 firebrick, closely stacked, occupy 56 cu. ft.
From 400 to 600 lb. of fire clay or dry cement (high temperature) are required to lay 1000 brick
About 135 lb. of plastic (unbaked) firebrick are required per cu. ft.

TABLE LIX.—STANDARD 9-IN. FIREBRICK SHAPES

37 STANDARD 9-IN. FIREBRICK SHAPES



9-IN. STRAIGHT
 $9 \times 4\frac{1}{2} \times 2\frac{1}{2}$ in.



**SMALL 9-IN.
BRICK**
 $9 \times 3\frac{1}{2} \times 2\frac{1}{2}$ in.



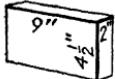
SOAP
 $9 \times 2\frac{1}{2} \times 2\frac{1}{4}$ in.



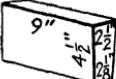
CHECKER
 $9 \times 2\frac{3}{4} \times 2\frac{3}{4}$ in.



SPLIT BRICK
 $9 \times 4\frac{1}{2} \times 1\frac{1}{4}$ in.



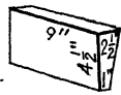
2-IN. BRICK
 $9 \times 4\frac{1}{2} \times 2$ in.



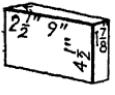
NO.1 ARCH
 $9 \times 4\frac{1}{2} \times (2\frac{1}{2} - 2\frac{1}{8})$ in.
76 brick to the circle
 $9 \times 4\frac{1}{2} \times (\frac{1}{2} - \frac{1}{4})$ in.
 $\frac{1}{4}$ ft. I.D.; 5 ft. O.D.



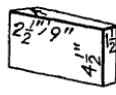
NO.2 ARCH
 $9 \times 4\frac{1}{2} \times (2\frac{1}{2} - 2\frac{1}{8})$ in.
76 brick to the circle
 $9 \times 4\frac{1}{2} \times (\frac{1}{2} - \frac{1}{4})$ in.
 $\frac{1}{4}$ ft. I.D.; $2\frac{1}{2}$ ft. O.D.



NO.3 ARCH
 $9 \times 4\frac{1}{2} \times (2\frac{1}{2} - 1)$ in.
19 brick to the circle
6 in. I.D.; $1\frac{1}{4}$ ft. O.D.



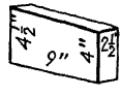
NO.1 WEDGE
 $9 \times 4\frac{1}{2} \times (2\frac{1}{2} - 1\frac{1}{8})$ in.
91 brick to the circle
 $4\frac{1}{2}$ ft. I.D.; 6 ft. O.D.



NO.2 WEDGE
 $9 \times 4\frac{1}{2} \times (2\frac{1}{2} - 1\frac{1}{2})$ in.
57 brick to the circle
 $2\frac{1}{4}$ ft. I.D.; $3\frac{3}{4}$ ft. O.D.



NO.3 WEDGE
 $9 \times 4\frac{1}{2} \times (3 - 2)$ in.
57 brick to the circle
 $3\frac{1}{2}$ ft. I.D.; $4\frac{1}{2}$ ft. O.D.



NO.1 KEY
 $9 \times (4\frac{1}{2} - 4) \times 2\frac{1}{2}$ in.
113 brick to the circle
12 ft. I.D.; $13\frac{1}{2}$ ft. O.D.



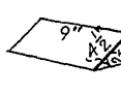
NO.2 KEY
 $9 \times (4\frac{1}{2} - 3\frac{1}{2}) \times 2\frac{1}{2}$ in.
57 brick to the circle
 $5\frac{1}{4}$ ft. I.D.; $6\frac{3}{4}$ ft. O.D.



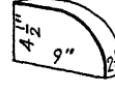
NO.3 KEY
 $9 \times (4\frac{1}{2} - 3) \times 2\frac{1}{2}$ in.
38 brick to the circle
3 ft. I.D.; $4\frac{1}{2}$ ft. O.D.



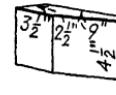
NO.4 KEY
 $9 \times (4\frac{1}{2} - 2\frac{1}{4}) \times 2\frac{1}{2}$ in.
25 brick to the circle
3 ft. I.D.; $4\frac{1}{2}$ ft. O.D.



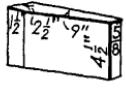
**FEATHER
EDGE**
 $9 \times 4\frac{1}{2} \times (2\frac{1}{2} - \frac{1}{8})$ in.



**JAMB
BRICK**
 $9 \times 4\frac{1}{2} \times 2\frac{1}{2}$ in.



NO.1 NECK
 $9 \times 4\frac{1}{2} \times 2\frac{1}{2} \times 3\frac{1}{2} \times \frac{5}{8}$ in.



NO.2 NECK
 $9 \times 4\frac{1}{2} \times 2\frac{1}{2} \times 1\frac{1}{2} \times \frac{5}{8}$ in.

TABLE LIX.—STANDARD 9-IN. FIREBRICK SHAPES.—(Continued)

37 STANDARD 9-IN. FIREBRICK SHAPES (CONTINUED)

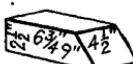
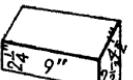
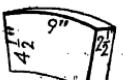
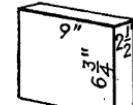
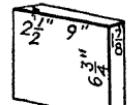
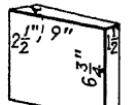
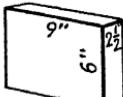
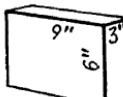
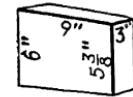
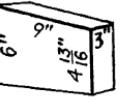
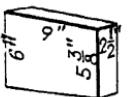
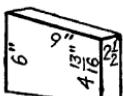
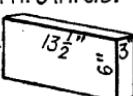
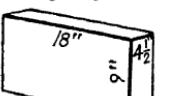
			
NO. 3 NECK $9 \times 4\frac{1}{2} \times (2\frac{1}{2}\frac{5}{8})$ in. $(9-6\frac{3}{4}) \times 4\frac{1}{2} \times 2\frac{1}{2}$ in.	END SKEW $9 \times (4\frac{1}{2}-2\frac{1}{4}) \times 2\frac{1}{2}$ in.	SIDE SKEW $9 \times (4\frac{1}{2}-1\frac{1}{2}) \times 2\frac{1}{2}$ in.	EDGE SKEW $9 \times (4\frac{1}{2}-1\frac{1}{2}) \times 2\frac{1}{2}$ in.
			
CIRCLE BRICK $9 \times 2\frac{1}{2} \times 4\frac{1}{2}$ in. 12 to 45 brick to the circle 24 to 20 in. I.D.; 33 to 129 in. O.D.	LARGE 9-IN. STRAIGHT $9 \times 6\frac{3}{4} \times 2\frac{1}{2}$ in. 91 brick to the circle	LARGE 9-IN. NO. 1 WEDGE $9 \times 6\frac{3}{4} \times (2\frac{1}{2}-1\frac{7}{8})$ in. $4\frac{1}{2}$ ft. I.D.; 6 ft. O.D.	LARGE 9-IN. NO. 2 WEDGE $9 \times 6\frac{3}{4} \times (2\frac{1}{2}-1\frac{1}{2})$ in. 57 brick to the circle $2\frac{1}{4}$ ft. I.D.; $3\frac{3}{4}$ ft. O.D.
			
FLAT BACK STRAIGHT $9 \times 6 \times 2\frac{1}{2}$ in. 2 1/2 ft. I.D.; 3 1/2 ft. O.D.	NO. 1 FLAT BACK ARCH $9 \times 6 \times (3\frac{1}{2}-2\frac{1}{2})$ in. $1\frac{1}{3}$ ft. I.D.; 2 1/2 ft. O.D.	NO. 2 FLAT BACK ARCH $9 \times 6 \times (3\frac{1}{2}-2)$ in. $1\frac{1}{3}$ ft. I.D.; 2 1/2 ft. O.D.	9x4 1/2 x 3 in. STRAIGHT
			
9X6X3-IN. STRAIGHT Also $9 \times 6 \times 2\frac{1}{2}$ in.	9x6x3 IN. NO. 2 KEY 9x($6-5\frac{3}{8}$)x3 in. 91 brick to the circle 12 ft. 11 in. I.D.; 14 ft. 5 in. O.D.	9x6x3 NO. 2 KEY 9x($6-4\frac{13}{16}$)x3 in. 48 brick to the circle 6 ft. 1 in. I.D.; 7 ft. 7 in. O.D.	9x6x2 1/2 IN. NO. 1 KEY 9x($6-5\frac{3}{8}$)x2 1/2 in. 91 brick to the circle 12 ft. 11 in. I.D.; 14 ft. 5 in. O.D.
		TWO SPECIAL SIZES	
9x6x2 1/2 IN. NO. 2 KEY 9x($6-4\frac{13}{16}$)x2 1/2 in. 48 brick to the circle 6 ft. 1 in. I.D.; 7 ft. 7 in. O.D.	13 1/2 IN. STRAIGHT $13\frac{1}{2} \times 6 \times 3$ in. Also $13\frac{1}{2} \times 6 \times 2\frac{1}{2}$ in.	BOTTOM BLOCK $18 \times 9 \times 4\frac{1}{2}$ IN.	

TABLE LX.—CHEMICAL ANALYSES OF FIRE-CLAY BRICKS, PER CENT*

Kind of brick	SiO ₂	Al ₂ O ₃	Fe ₂ O ₃	P.C.E.†
New Jersey..	79.8	19.0	1.2	30
New Jersey..	77.8	20.4	2.0	30–31
Pennsylvania	51.2	43.9	2.8	32–33
Pennsylvania	53.9	41.9	1.9	33
Kentucky....	61.4	34.3	2.4	30
Missouri.....	48.2	48.0	1.8	33
Missouri.....	55.4	41.2	1.5	32–33

* From Marks, "Mechanical Engineers' Handbook," McGraw-Hill Book Company, Inc.

† P.C.E. is defined by Committee C-8 of the A.S.T.M. as "an index to the degree of fusion resulting in a cone of the material bending until the tip touches the plaque as a result of a definite heat treatment." For the equivalent of cones in temperature see Table LXI.

TABLE LXI.—FUSING POINTS OF SEGER CONES*

Number of cone	Fusing point original scale		Number of cone	Fusing point original scale		Revised scale†	
	°F.	°C.		°F.	°C.	°F.	°C.
.022	1,094	590	10	2,426	1,330		
.021	1,148	620	11	2,462	1,350		
.020	1,202	650	12	2,498	1,370		
.019	1,256	680	13	2,534	1,390		
.018	1,310	710	14	2,570	1,410		
.017	1,364	740	15	2,606	1,430		
.016	1,418	770	16	2,642	1,450		
.015	1,472	800	17	2,678	1,470		
.014	1,526	830	18	2,714	1,490	2,714	1,490
.013	1,580	860	19	2,750	1,510	2,750	1,510
.012	1,634	890	20	2,786	1,530	2,786	1,530
.011	1,688	920	21‡	2,822	1,550		
.010	1,742	950	22‡	2,858	1,570		
.09	1,778	970	23‡	2,894	1,590		
.08	1,814	990	24‡	2,930	1,610		
.07	1,850	1,010	25‡	2,966	1,630		
.06	1,886	1,030	26	3,002	1,650	2,912	1,600
.05	1,922	1,050	27	3,038	1,670	2,948	1,620
.04	1,958	1,070	28	3,074	1,690	2,975	1,635
.03	1,994	1,090	29	3,110	1,710	3,002	1,650
.02	2,030	1,110	30	3,146	1,730	3,038	1,670
.01	2,066	1,130	31	3,182	1,750	3,065	1,685
1	2,102	1,150	32	3,218	1,770	3,101	1,705
2	2,138	1,170	33	3,254	1,790	3,128	1,720
3	2,174	1,190	34	3,290	1,810	3,164	1,740
4	2,210	1,210	35	3,326	1,830	3,191	1,755
5	2,246	1,230	36	3,362	1,850		
6	2,282	1,250	37	3,398	1,870		
7	2,318	1,270	38	3,434	1,890		
8	2,354	1,290	39	3,470	1,910		
9	2,390	1,310					

* From Hauck's "Industrial Combustion Data."

† U.S. Bureau of Standards, Washington, D. C.

‡ Cones 21 to 25 inclusive, all come down at practically the same temperature.

TABLE LXII.—CONTENTS OF HORIZONTAL TANKS*
Multiply Factor in Tables by Length of Tank in Inches to Get Gallons

48 in. in diameter	49 in. in diameter	50 in. in diameter	Depth, in.	51 in. in diameter	52 in. in diameter	53 in. in diameter
.....	26½	4.776
.....	26	4.597	4.660
.....	25½	4.422		
.....	4.250	4.250	25	4.309	4.371	4.431
.....	4.082	24½			
3.917	3.975	4.033	24	4.085	4.146	4.203
3.707	3.765	3.817	23	3.865	3.922	3.976
3.498	3.555	3.602	22	3.647	3.700	3.749
3.289	3.345	3.388	21	3.431	3.479	3.523
3.084	3.136	3.175	20	3.216	3.259	3.300
2.881	2.928	2.964	19	3.002	3.044	3.078
2.679	2.722	2.755	18	2.790	2.825	2.859
2.478	2.517	2.548	17	2.580	2.613	2.644
2.281	2.316	2.344	16	2.374	2.405	2.423
2.087	2.118	2.145	15	2.170	2.199	2.222
1.900	1.924	1.948	14	1.971	1.996	2.016
1.716	1.734	1.756	13	1.777	1.797	1.815
1.533	1.550	1.509	12	1.585	1.605	1.622
1.353	1.370	1.386	11	1.402	1.417	1.433
1.180	1.195	1.210	10	1.223	1.235	1.251
1.017	1.027	1.040	9	1.052	1.063	1.077
0.859	0.866	0.878	8	0.888	0.897	0.907
0.708	0.716	0.723	7	0.729	0.737	0.746
0.565	0.575	0.578	6	0.583	0.587	0.595
0.432	0.440	0.442	5	0.447	0.451	0.454
0.310	0.317	0.319	4	0.319	0.326	0.329
0.201	0.205	0.208	3	0.211	0.214	0.214
0.133	0.114	0.114	2	0.114	0.117	0.119
0.040	0.041	0.041	1	0.041	0.041	0.042

* From Cross, "Handbook of Petroleum, Asphalt and Natural Gas," Kansas City Testing Laboratory.

TABLE LXII.—CONTENTS OF HORIZONTAL TANKS.*—(Continued)

54 in. in diameter	55 in. in diameter	56 in. in diameter	Depth, in.	57 in. in diameter	58 in. in diameter	59 in. in diameter
.....	29½	5.918
.....	29	5.719	5.790
.....	28½	5.523		
.....	5.331	28	5.390	5.467	5.535
.....	5.143	27½			
4.957	5.023	5.089	27	5.153	5.217	5.280
4.723	4.785	4.847	26	4.907	4.967	5.026
4.490	4.547	4.605	25	4.662	4.717	4.773
4.258	4.311	4.365	24	4.417	4.469	4.521
4.026	4.076	4.125	23	4.175	4.223	4.271
3.794	3.842	3.886	22	3.934	3.987	4.023
3.566	3.611	3.651	21	3.694	3.736	3.777
3.340	3.381	3.418	20	3.456	3.495	3.534
3.116	3.152	3.188	19	3.222	3.256	3.293
2.893	2.926	2.950	18	2.992	3.020	3.057
2.674	2.704	2.734	17	2.766	2.788	2.823
2.459	2.486	2.513	16	2.543	2.563	2.594
2.248	2.271	2.296	15	2.321	2.344	2.369
2.041	2.061	2.084	14	2.104	2.128	2.149
1.838	1.857	1.878	13	1.893	1.916	1.934
1.640	1.657	1.675	12	1.692	1.710	1.726
1.449	1.464	1.478	11	1.496	1.509	1.524
1.265	1.279	1.290	10	1.304	1.316	1.329
1.086	1.099	1.108	9	1.120	1.130	1.141
0.915	0.926	0.936	8	0.943	0.953	0.961
0.755	0.759	0.769	7	0.776	0.784	0.791
0.602	0.607	0.614	6	0.620	0.626	0.631
0.461	0.466	0.470	5	0.473	0.479	0.483
0.331	0.335	0.337	4	0.340	0.344	0.347
0.217	0.219	0.220	3	0.223	0.225	0.227
0.119	0.120	0.121	2	0.122	0.123	0.124
0.042	0.042	0.043	1	0.043	0.044	0.044

* From Cross, "Handbook of Petroleum, Asphalt and Natural Gas," Kansas City Testing Laboratory.

TABLE LXII.—CONTENTS OF HORIZONTAL TANKS.*—(Continued)

60 in. in diameter	61 in. in diameter	62 in. in diameter	Depth, in.	63 in. in diameter	64 in. in diameter	65 in. in diameter
.....	32½	7.182
.....	32	6.963	7.039
.....	31½	6.747		
.....	6.535		31	6.610	6.686	6.755
.....	6.326	30½			
6.119	6.193	6.267	30	6.337	6.410	6.472
5.858	5.929	5.999	29	6.065	6.134	6.193
5.598	5.668	5.732	28	5.794	5.858	5.915
5.339	5.407	5.465	27	5.523	5.584	5.639
5.082	5.146	5.199	26	5.254	5.310	5.363
4.826	4.885	4.935	25	4.986	5.038	5.089
4.572	4.625	4.672	24	4.722	4.771	4.817
4.318	4.366	4.412	23	4.458	4.503	4.547
4.066	4.111	4.153	22	4.196	4.239	4.281
3.818	3.859	3.898	21	3.937	3.976	4.016
3.572	3.609	3.645	20	3.683	3.718	3.756
3.328	3.363	3.397	19	3.430	3.464	3.496
3.088	3.120	3.151	18	3.181	3.213	3.242
2.852	2.881	2.910	17	2.937	2.964	2.992
2.621	2.646	2.672	16	2.698	2.723	2.748
2.392	2.417	2.440	15	2.463	2.486	2.508
2.171	2.192	2.213	14	2.232	2.254	2.274
1.954	1.972	1.991	13	2.008	2.027	2.045
1.743	1.759	1.776	12	1.791	1.808	1.823
1.538	1.552	1.567	11	1.581	1.595	1.608
1.341	1.352	1.366	10	1.378	1.390	1.401
1.152	1.161	1.173	9	1.183	1.192	1.203
0.971	0.980	0.988	8	0.906	1.005	1.013
0.799	0.806	0.812	7	0.819	0.827	0.833
0.634	0.642	0.648	6	0.653	0.659	0.664
0.487	0.491	0.496	5	0.500	0.504	0.506
0.349	0.354	0.357	4	0.359	0.362	0.365
0.229	0.230	0.233	3	0.235	0.238	0.238
0.125	0.126	0.128	2	0.128	0.129	0.131
0.045	0.045	0.045	1	0.046	0.046	0.047

* From Cross, "Handbook of Petroleum, Asphalt and Natural Gas," Kansas City Testing Laboratory.

TABLE LXII.—CONTENTS OF HORIZONTAL TANKS.*—(Continued)

66 in. in diameter	67 in. in diameter	68 in. in diameter	Depth, in.	69 in. in diameter	70 in. in diameter	71 in. in diameter
.....	35½	8.570
.....	35	8.330	8.413
.....	34½	8.094		
.....	7.861	7.861	34	7.944	8.026	8.107
.....	7.631	33½			
7.406	7.485	7.567	33	7.640	7.723	7.801
7.120	7.194	7.273	32	7.348	7.421	7.495
6.834	6.904	6.979	31	7.051	7.120	7.190
6.549	6.617	6.687	30	6.755	6.819	6.886
6.264	6.327	6.395	29	6.459	6.519	6.583
5.981	6.041	6.104	28	6.164	6.222	6.283
5.699	5.756	5.814	27	5.870	5.927	5.983
5.419	5.473	5.528	26	5.580	5.634	5.686
5.141	5.191	5.244	25	5.292	5.343	5.391
4.865	4.913	4.961	24	5.006	5.052	5.098
4.592	4.637	4.681	23	4.724	4.764	4.809
4.322	4.363	4.403	22	4.444	4.481	4.524
4.054	4.092	4.128	21	4.167	4.204	4.241
3.789	3.824	3.859	20	3.893	3.929	3.962
3.529	3.561	3.593	19	3.625	3.657	3.688
3.273	3.302	3.331	18	3.360	3.388	3.418
3.020	3.046	3.074	17	3.101	3.125	3.152
2.772	2.797	2.821	16	2.846	2.868	2.894
2.530	2.553	2.575	15	2.595	2.617	2.640
2.294	2.314	2.333	14	2.352	2.372	2.391
2.064	2.080	2.099	13	2.116	2.135	2.150
1.839	1.855	1.871	12	1.886	1.901	1.916
1.622	1.635	1.650	11	1.663	1.674	1.693
1.413	1.426	1.439	10	1.449	1.459	1.476
1.213	1.223	1.235	9	1.242	1.254	1.264
1.022	1.030	1.041	8	1.047	1.060	1.063
0.841	0.847	0.855	7	0.859	0.871	0.874
0.670	0.675	0.680	6	0.687	0.689	0.697
0.512	0.516	0.529	5	0.524	0.528	0.531
0.368	0.371	0.374	4	0.377	0.378	0.382
0.240	0.243	0.244	3	0.246	0.249	0.250
0.131	0.132	0.133	2	0.134	0.135	0.136
0.047	0.047	0.047	1	0.048	0.048	0.048

* From Cross, "Handbook of Petroleum, Asphalt and Natural Gas," Kansas City Testing Laboratory.

TABLE LXII.—CONTENTS OF HORIZONTAL TANKS.*—(Continued)

72 in. in diameter	73 in. in diameter	74 in. in diameter	Depth, in.	75 in. in diameter	76 in. in diameter	77 in. in diameter
.....	38½	10.079
.....	38	9.819	9.912
.....	37½	9.562		
.....	9.309	37	9.400	9.489	9.579
.....	9.059	36½			
8.813	8.899	8.989	36	9.076	9.160	9.246
8.500	8.582	8.669	35	8.752	8.832	8.914
8.188	8.267	8.349	34	8.428	8.505	8.583
7.887	7.953	8.030	33	8.104	8.178	8.253
7.567	7.639	7.712	32	7.782	7.852	7.924
7.259	7.326	7.395	31	7.461	7.528	7.596
6.952	7.015	7.080	30	7.142	7.205	7.268
6.645	6.706	6.766	29	6.824	6.885	6.944
6.341	6.397	6.454	28	6.509	6.567	6.622
6.038	6.091	6.145	27	6.195	6.250	6.302
5.736	5.786	5.839	26	5.885	5.938	5.988
5.439	5.485	5.535	25	5.578	5.628	5.675
5.144	5.188	5.232	24	5.274	5.320	5.364
4.852	4.892	4.934	23	4.975	5.014	5.056
4.563	4.599	4.639	22	4.677	4.715	4.753
4.278	4.311	4.374	21	4.383	4.418	4.453
3.997	4.025	4.062	20	4.094	4.127	4.161
3.719	3.748	3.781	19	3.809	3.839	3.871
3.446	3.474	3.501	18	3.529	3.556	3.585
3.179	3.204	3.229	17	3.255	3.280	3.305
2.917	2.938	2.962	16	2.985	3.008	3.032
2.658	2.681	2.702	15	2.723	2.744	2.764
2.408	2.429	2.447	14	2.467	2.485	2.503
2.167	2.184	2.200	13	2.216	2.234	2.250
1.932	1.946	1.960	12	1.978	1.990	2.003
1.703	1.716	1.727	11	1.742	1.753	1.767
1.483	1.494	1.505	10	1.515	1.527	1.538
1.272	1.281	1.291	9	1.300	1.309	1.318
1.071	1.079	1.086	8	1.095	1.102	1.110
0.880	0.887	0.893	7	0.899	0.906	0.912
0.701	0.707	0.712	6	0.717	0.722	0.727
0.536	0.540	0.544	5	0.548	0.551	0.555
0.386	0.388	0.391	4	0.393	0.396	0.399
0.252	0.253	0.254	3	0.256	0.259	0.260
0.138	0.138	0.139	2	0.140	0.141	0.142
0.048	0.049	0.049	1	0.050	0.050	0.050

* From Cross, "Handbook of Petroleum, Asphalt and Natural Gas," Kansas City Testing Laboratory.

TABLE LXIII.—OVER-ALL THERMAL CONDUCTIVITIES FOR HEAT EXCHANGERS*

Values of K , the over-all coefficient of conductivity, for the equation

$$Q = K\sqrt{VA}d$$

In transferring heat from

water to condensing vapors.....	20- 35
water to oil.....	12- 20
oil to condensing vapors.....	11- 17
oil to oil.....	8- 12
water to condensing steam.....	200-400
oil to steam.....	25- 50

* From BELL, "American Petroleum Refining," D. Van Nostrand Company, Inc., 1930.

TABLE LXIV.—FUNDAMENTAL DATA FOR BOILER AND HEATING-PLANT DESIGN

The heat of evaporation of water, from and at 212°F., is 970.4 B.t.u. per lb.

One sq. ft. of steam radiation emits 242.6 B.t.u. per hr. to surrounding atmosphere when the environment is at 70°F. and the radiator is filled with steam at 2 lb. pressure gauge

One sq. ft. of hot-water radiation emits 150 B.t.u. per hr. under ordinary conditions of operation

A boiler horsepower is defined as the equivalent of the evaporation of 34.5 lb. of water per hr. from and at 212°F. Hence 1 boiler hp. equals $34.5 \times 970.4 = 33,479$ B.t.u. per hr.

It follows that 1 boiler hp. is equivalent to 138 sq. ft. of steam radiation, and to 223 sq. ft. of hot-water radiation

Boiler heating surface is defined as boiler surface having water on one side and hot combustion gases on the other. Nominally, a boiler horsepower is equivalent to 10 sq. ft. of boiler heating surface, but many modern boilers develop a horsepower with much less surface

TABLE LXV.—DATA ON METALLURGICAL OIL-FIRED FURNACES*

The required furnace volumes for fuel-oil fires in various furnaces are:

Temp. of Operation, °F.	B.t.u. Liberated per Cu. Ft. of Combustion	Volume per Hour
600-1500		1,000- 15,000
1500-1800		15,000- 22,000
1800-2000		22,000- 35,000
2000-3000		35,000-150,000
Operation	Approximate temp. range,	Approximate quantity of fuel required, B.t.u.
Steel-mill operations		Per gross ton
Bar and shape furnaces.....	1900-2200	4,500,000
Bolt heading.....	2200-2300	5,600,000
Flanging.....	1800-2100	6,000,000
Ingot heating.....	2000-2200	3,000,000- 3,500,000
Nail bluing.....	650	1,000,000
Nail galvanizing.....	800- 900	600,000
Open hearth.....	2800-3100	4,000,000- 7,000,000
Pipe butt welding.....	2400-2600	5,000,000
Pipe lap welding, bend and weld—2 operations	1800-2600	6,000,000
Pipe galvanizing.....	800- 900	2,250,000
Pot annealing wire.....	1650	4,000,000
Puddle-mill heating.....		14,000,000-20,000,000
Rail-bloom reheating, hot blooms.....	1900-2050	2,000,000
Rod-mill furnaces.....	1900-2100	1,400,000
Sheet galvanizing.....	800- 900	1,250,000
Sheet-jobbing mill.....	2000-2100	5,000,000
Sheet-mill furnaces.....	1800-2100	2,750,000
Sheet-box annealing.....	1650	2,800,000
Sheet-blue annealing.....	1400-1600	3,500,000
Slab heating.....	2000-2200	3,000,000- 3,500,000
Soaking pits.....	1900-2100	1,000,000- 1,500,000
Spike heating.....	2000-2200	5,600,000
Tin-plate box annealing.....	1200-1650	1,250,000
Tin-plate hot mills.....	1800-2000	3,000,000
Tin-plating sheets.....	650	1,125,000
Wire baking.....	300- 350	
Wire galvanizing.....	800- 900	4,000,000
Metal-working plants		Per lb.
Annealing high-carbon steel.....	1400-1500	
Annealing springs.....	1500-1650	1000
Annealing wire.....	1200-1400	
Carburizing.....	1400-1700	
Drop forging.....	2200-2300	2000
Hardening high-speed steel.....	1800-2200	
Hardening tools.....	1400-2200	1250
Heat treating.....	700-1650	1050
Lead and cyanide hardening.....	1400-1800	
Oil tempering.....	500	
Rivet heating.....	2000-2200	
Tempering.....	425- 650	
Welding.....	2100-2800	

* The data of this table are given by Hauck in "Industrial Combustion Data."

TABLE LXV.—DATA ON METALLURGICAL OIL-FIRED FURNACES.*—
(Continued)

Operation	Approximate temp. range,	Approximate quantity of fuel required, B.t.u.
Iron and steel foundries		Per lb.
Core baking.....	350-600	
Annealing malleable castings.....	1500-1700	
Annealing steel castings.....	1300-1600	
Nonferrous metal operations		
Melting aluminum.....	1217	
Melting babbitt.....	600- 800	
Melting brass.....	1650-2250	2650
Melting bronze.....	1800-2300	2650
Refining copper.....	2100-2600	
Melting electrolyte.....	700	
Melting lead.....	620	
Melting monel metal.....	2800	
Melting nickel.....	2650	
Melting solder.....	400	
Melting stereotype.....	700	
Annealing aluminum.....	390- 850	
Annealing brass.....	475-1000	
Annealing copper.....	450- 950	
Annealing monel metal.....	1100-1480	
Annealing nickel.....	1100-1480	
Ceramic operations		
Burning brick.....	2500-2800	6000/brick
Glass annealing.....	650-1500	520
Glass making.....	2600-3000	10,500
Glazing china.....	1500-1900	
Porcelain burning.....	2600	
Vitreous enameling.....	1400-1800	
Other operations		
Bread baking.....	300-500	
Cement kilns.....	2800-3000	1,650,000/bbl.
Coffee roasting.....	600-700	5000
Calcining pigments.....	1600	
Asphalt melting.....	350-450	
Lithographing.....	300	
Japanning.....	300-450	
Enameling.....	250-450	
Pie baking.....	500	
Varnish cooking.....	550	

TABLE LXVI.—COEFFICIENTS OF CUBICAL EXPANSION*
 Mean values of 1000α at ordinary room temperatures for use in the
 equation $V_2 = V_1 + \alpha V_1(t_2 - t_1)$

Liquids	1000α	Solids	1000α
Acetic acid.....	0.80	Fluorspar.....	0.035
Alcohol (ethyl).....	0.61	Ice (4 to 30°F.).....	0.62
Alcohol (methyl).....	0.80	Paraffin wax.....	0.61
Benzene.....	0.77	Rock salt.....	0.67
Benzol.....	0.70	Sulphur.....	0.40
Calcium chloride (CaCl_2); 5 to 50% solution.....	0.28	Wood (Beech).....	0.016
Chloroform.....	0.77	Wood (Pine).....	0.028
Ether.....	0.92		
Glycerine.....	0.28		
Hydrochloric acid.....	0.27		
Hydrochloric acid (50% solu- tion).....	0.52		
Olive oil.....	0.41		
Petroleum, Pennsylvania.....	0.50		
Petroleum, California.....	0.43		
Petroleum, Texas.....	0.42		
Phenol ($\text{C}_6\text{H}_5\text{O}$).....	0.50		
Rape-seed oil.....	0.50		
Salt, 1.6% solution.....	0.60		
Salt, 26% solution.....	0.24		
Sulphuric acid.....	0.31		
Sulphuric acid, 50% solution	0.45		
Turpentine.....	0.54		
Water.....	0.115		

* From Marks, "Mechanical Engineers' Handbook," McGraw-Hill Book Company, Inc.

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